JUN 8 - 1937

Heating · Piping and Air Conditioning

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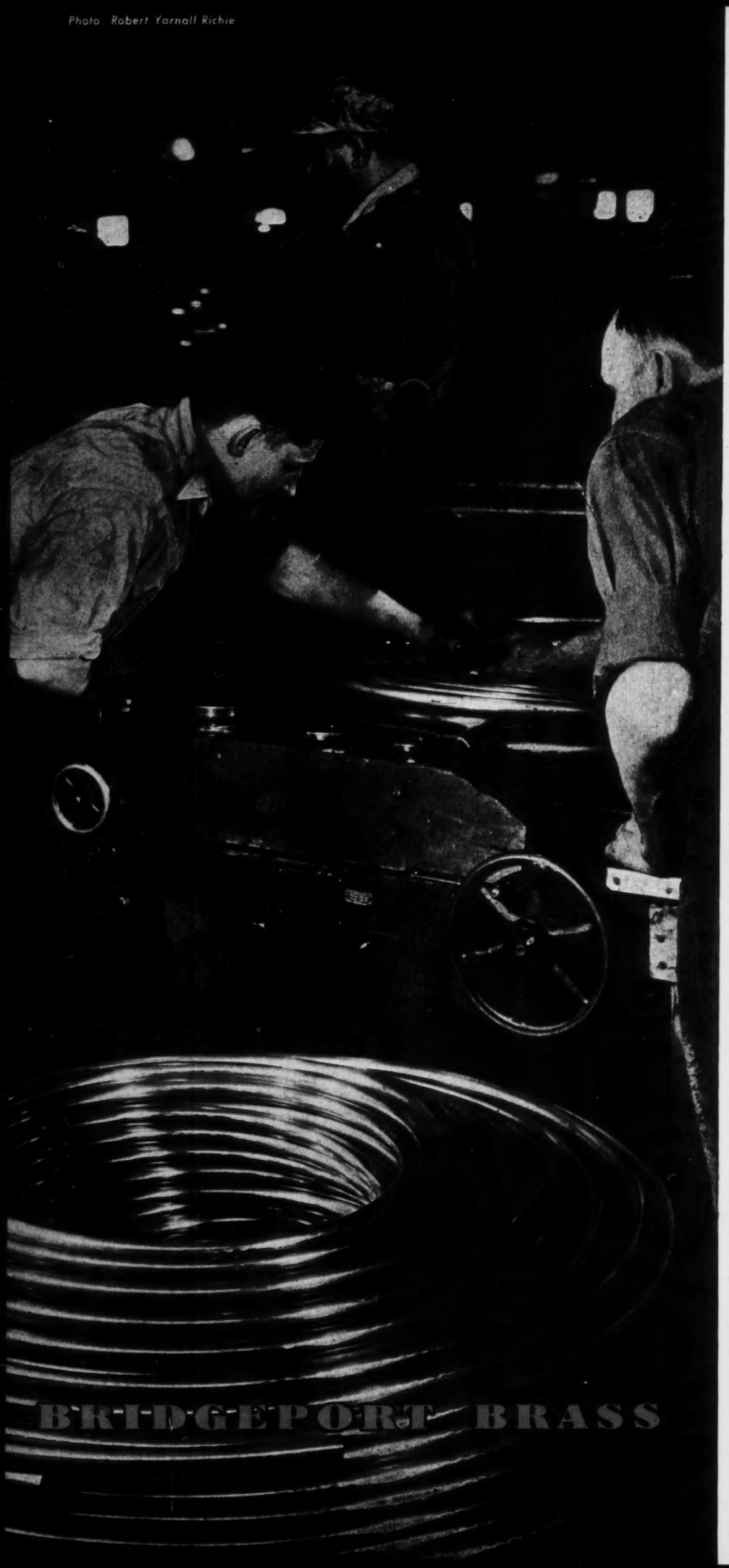

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The Editor's Page

■ **Installed cost** of equipment sold by the members of the Air Conditioning Manufacturers' Association broke all records in the first four months of 1937, aggregating $41,311,301, increase of 180 per cent over the sales of $14,756,992 in the same months a year ago, and a total not reached in 1936 until October, according to William B. Henderson. executive vice-president. . . . April sales were second highest of any month in history, being $9,663,009, an increase of 156.8 per cent over the April, 1936, total, which was $3,751,015. . . . The Association's sales in 1936, biggest year of the industry's history, were $49,942,301, and the sales reported this year already are 82.7 per cent of that total, Mr. Henderson said. The seasonal spring peak was reached in March, with sales of $17,137,869. . . . J. A. Harlan, Kelvinator executive, was elected president of the Association at the annual meeting at Hot Springs, Va., last month. William H. Price, Jr., vice-president, Carrier Corp., was made vice-president, and P. A. McKittrick, general manager, Parks-Cramer Co., treasurer. Directors are the officers and W. F. Armstrong, assistant general manager, Delco-Frigidaire Conditioning Div.; P. Y. Danley, manager, Refrigeration and Air Conditioning Dept., Westinghouse Electric & Mfg. Co.; J. J. Donovan, manager, Air Conditioning Dept., General Electric Co.; Stewart E. Lauer, vice-president, York Ice Machinery Corp.; and J. F. G. Miller, vice-president, B. F. Sturtevant Co.

■ **Plant construction** for the air conditioning and kindred industries is only now beginning to hit its stride, according to George A. Bryant, Jr., executive vice-president of the Austin Co. While many of the projects are small the care taken to insure opportunity for manifold expansion indicates the outlook for this field, says Mr. Bryant. . . . With industrial construction awards continuing near the levels set during the first four months of this year and no decrease in the requirements for new plant, factory office, or research facilities apparent continuance of the present high rate of plant construction through the summer and fall is probable, he believes. . . . 35 per cent more air conditioning installations were made in Philadelphia during the first four months of this year than in the entire twelve months of 1936, according to the "Air Conditioning News" published by the Philadelphia Electric Co. . . . A definite warning note to all establishments not yet air conditioned, says the utility. . . . The Refrigeration and Air Conditioning Institute, Chicago, has completed plans for the construction of a new building unit to cost an estimated $50,000, which will be of brick and steel construction with windowless walls of glass brick, air conditioned throughout. . . . The new unit was decided upon because of the steadily increasing growth of air conditioning and the consequent increased demand for trained men. . . . The Gay Engineering Corp., Los Angeles, a pioneer air conditioning firm of southern California, has utilized glass blocks for its new air conditioned office and industrial building. . . . The same refrigerating compressors used in summer for cooling the building warm it in winter. . . . Because the furniture buyer's market occurs during June and July, the Imperial Furniture Co., Grand Rapids, Mich., has erected a new air conditioned windowless show room building, the first in the city to be exclusively erected for air conditioning. Air is cooled to 52 F by water from specially drilled wells in separate units on each floor.

■ **The Palmer House,** famous Chicago hotel, is installing the first of the air conditioning equipment which ultimately will comprise a $1,000,000 air conditioning program and which will be extended to include 2248 guest rooms, six public dining rooms, three bars, 20 private meeting and dining rooms, two ball rooms, and more than 60 stores and shops, according to Edward T. Lawless, general manager. The space to be air conditioned will cover 21,500,000 cu ft. . . . This year's efforts will see the grand ball room, the Red Lacquer room, and their attendant foyers, the exhibition hall, the 20 private dining rooms, a number of guest rooms, and the street level stores and shops air conditioned. . . . Charles L. Leopold is the consulting engineer. . . . A New York hotel advertises air conditioning as the "Fourth Dimension" in hotel rooms.

■ **A plan** of medical and engineering research to combat industrial diseases due to dust and fumes has been announced by the Air Hygiene Foundation of America, Inc., 4400 Fifth Ave., Pittsburgh. The program calls for studies at several laboratories and institutions, and the work of the Foundation is being supported by some 170 different firms. . . . According to the Foundation, industrial diseases of this kind are a potential peril to the health of more than one million workmen. The research will seek to discover: (1) Basic knowledge essential to determine the cause and prevention of diseases attributable to dusts and other industrial air pollutants. (2) What the employer must do to safeguard the health of his workmen and to protect himself against false damage claims. (3) Curtailment of the health hazards inherent in silica. (4) More practical information in the field of chest x-ray. (5) How to reduce the high pneumonia rate in smoky atmospheres. . . . The Industrial Unit Heater Association, 5-208 General Motors Bldg., Detroit, has made available for distribution the "Standard Code for Testing and Rating Steam Unit Heaters" prepared by a joint committee of the Association and the American Society of Heating and Ventilating Engineers. A copy of the publication may be obtained upon written request to the Association.

■ **The month after** the installation of an air conditioning apparatus in his dental office, R. L. Lasater treated more patients than any other month in that year; consequently Dr. Lasater says he has little hesitancy in saying he can see air conditioning as a factor that may aid overcoming seasonal slumps, long a bug-bear to dentists. Besides pleasing the patients, there is no doubt that air conditioning has increased the efficiency of the dentists. . . . The type of air conditioning installed and these and its other advantages are described in an article on weather control of the dental office in the May issue of "Oral Hygiene."

■ **Subjects** affecting standardization of underground piping systems are to be discussed at a conference on corrosion to be sponsored by the National Bureau of Standards during October or November, according to "Industrial Standardization and Commercial Standards Monthly" published by the American Standards Association. . . . Topics suggested which will have a bearing on standardization are: Methods for determining the corrosivity of soils, methods for determining the condition of pipe lines, and tests and specifications for pipe coatings. . . . Those wishing to take part in the conference should notify K. H. Logan, National Bureau of Standards, Washington, D. C.

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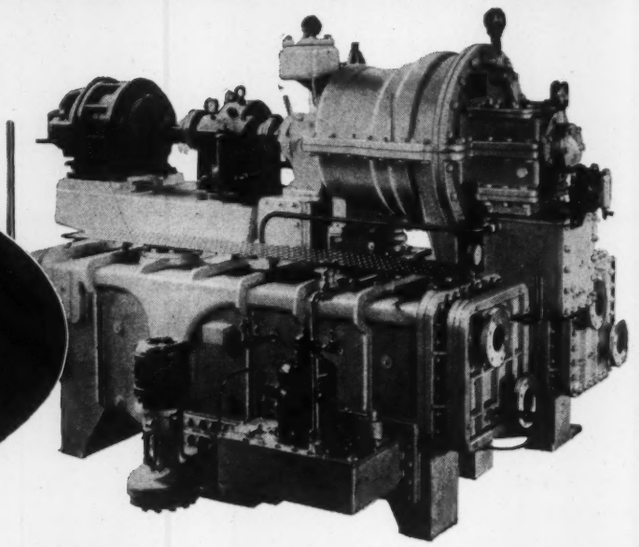
Carrier Centrifugal REFRIGERATION



Office Buildings, Banks and Industrial Plants among Hundreds Reporting Economical Temperature Control for Air Conditioning Use

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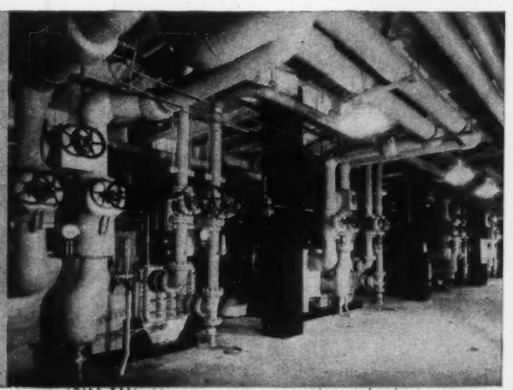
FEWER PARTS mean less wear. Writes Chief Engineer Davidson of the Northern Trust Company, Chicago, "We have found the machine so dependable and automatic in operation that it requires little attention, other than normal observation. Very few bearing parts—consequently the maintenance cost is nominal compared to other makes."

ADAPTABILITY is a big feature. In the Philadelphia Savings Fund Society Building, for example, a number of Carrier Centrifugals are combined to form a battery capable of removing 11,500,000 BTU per hour from the building.



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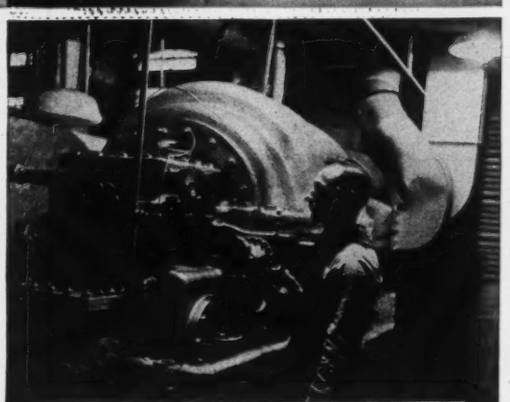

DEPENDABILITY is still another feature. For 12 years, two Carrier Centrifugals have provided the cooling for Health Products in Newark, N. J. Satisfactory? Ask Chief Engineer Smith!

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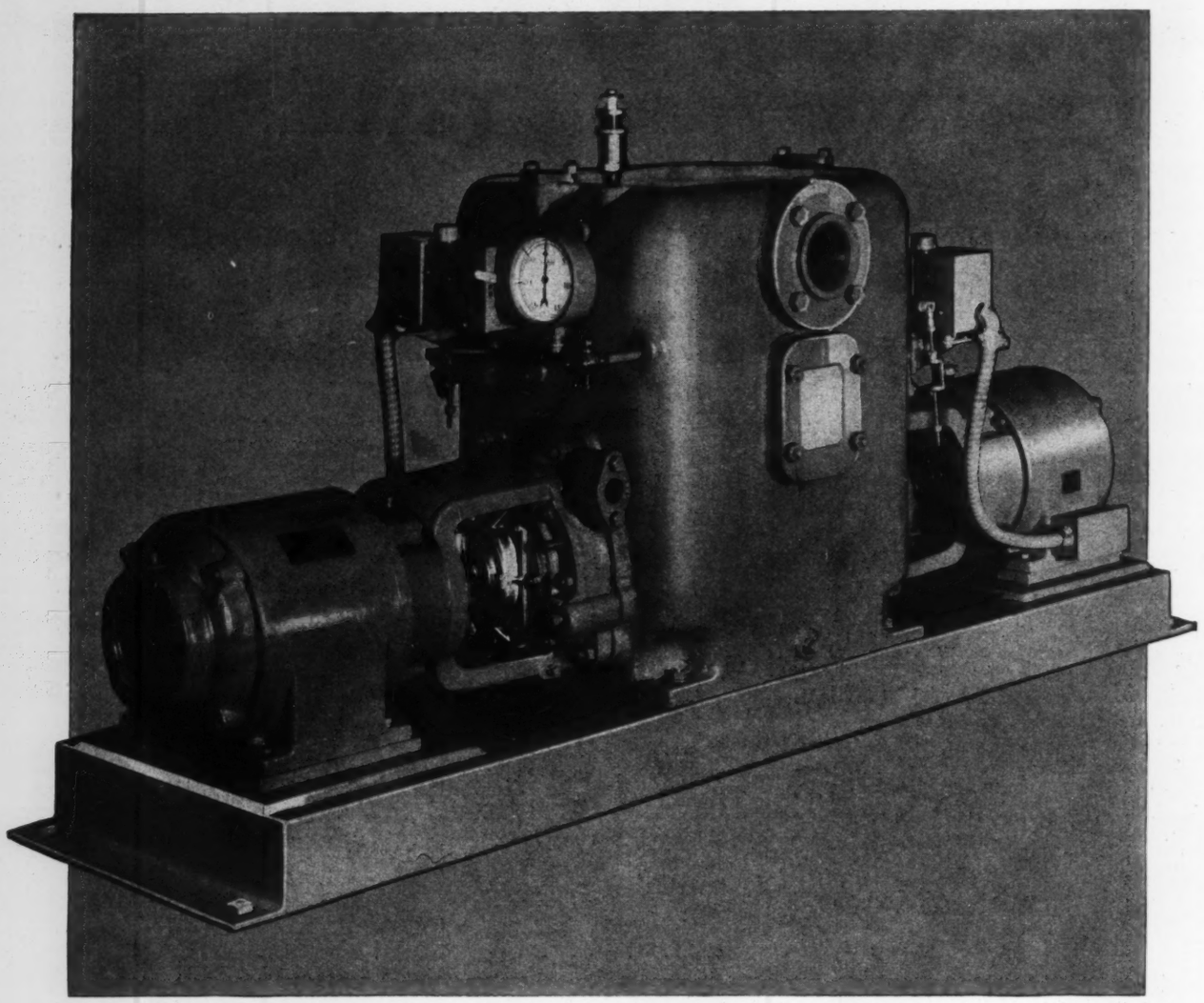

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Heating · Piping and Air Conditioning

Vol. 9, No. 6 • June, 1937

Rayon and Air Conditioning

How Rayon Is Made; Air Conditioning Requirements of the Process

By L. L. Lewis*

Modern air conditioning, generally acknowledged as born in the spinning room of a cotton mill some thirty years ago, is today essential to the manufacturing process of another branch of the textile industry —rayon. In fact, it has been said that there would be no rayon industry were it not for air conditioning. . . . To the National Rayon Technical Conference at Washington last month—commemorating the 25th anniversary of the manufacture of rayon in the United States—Mr. Lewis presented a comprehensive paper on air conditioning for textile plants making and using synthetic yarns. Because of the importance of ar conditioning to the process and as an example of what air conditioning means to modern industry, the part of his paper dealing with the process air conditioning requirements is published

CONSIDERATION of the problems of application of air conditioning to textile plants making and using synthetic yarns enforces a choice between two points of view: (1) that of grouping all problems under purpose classifications; and (2) that of following the flow of material through the process. The first seems to offer brevity to all, but clarity only to those having a clear picture of the processes of manufacture. The latter compels some repetition but is chosen because it appears to offer greater clarity to those less familiar with the details of process.

As to manufacturing processes, there are viscose, acetate, cupra-ammonium, nitro-cellulose, and others—some being widely used, others hardly at all. Each is different from the other and each presents its own set of problems in air conditioning—all of which are different from the problems in connection with the processing of the yarn in the plant which buys it from the manufacturer.

Considerations of some processes of yarn manufacture may be promptly discarded because of the small volume produced by a particular process. The order of discussion is, therefore, arbitrarily set as viscose, acetate, and the processing of yarns produced by each.

For the sake of clarity, reference is made to the outline of the flow of material in a viscose plant and the separate processes through which it will pass. Recognition must be given to the fact that improvements are constantly being made and that the experiences and developments of various manufacturers have led them to departures from this particular chart. Nevertheless, it represents a practical process, and the modifications existing in practice do not materially alter air conditioning application problems.

(*Note:* The author makes no claim to expertness—either as to chemistry or the actual production of rayon—and wishes to acknowledge with appropriate humility that his knowledge in these respects is a borrowed one. A considerable knowledge of the process is, however, essential to a comprehension of the air conditioning problems which are involved. His description of processes is, therefore, presented primarily for that purpose.)

Viscose Process

Step 1—Raw Material Storage. Commonly used raw materials are spruce pulp and cotton linters. While it is recognized that the regain of either of these materials will vary over a wide range as they are taken from unconditioned storage, the effect of this regain upon subsequent process is considered to be negligible. Depending upon the time of storage and the season of the year, this variation will undoubtedly reach the magnitude of 4 lb of water for each 100 lb of bone dry materials. With drying and conditioning equipment, the regain of this material could readily be fixed within very narrow limits. By conditioning the storage room, the amplitude of the peaks and valleys otherwise obtaining could be reduced, but whatever the effect may be upon subsequent processes, its magnitude is so small that the cost of controlling regain is unwarranted. Heating, only, is required for the storage space.

Steps 2 and 3—Steeping and Shredding. The sheets of cellulose raw material are gathered into batches of a given weight and submerged in a caustic soda solution. It is important that the entire batch be uniformly soaked with the alkali, for in this operation the soda combines with the wood pulp to form sodium cellulose. After complete impregnation, the soda solution is drained off and the excess soda in the sheets is removed by means of pressure.

In order thoroughly to mix the sodium cellulose, the sheets are shredded into snow white flocks or crumbs by means of machines, known variously as shredders or

*Vice-President and Chief Engineer, Carrier Corp.

"pfleiderers." Steeping and shredding are usually conducted in the same room and for this reason the processes are considered together. Steeping requires no air conditioning.

A considerable amount of power is required to drive the shredder and a considerable amount of heat is generated by the chemical reaction, and all of this heat must be absorbed to prevent an undesirable acceleration of the latter. The shredder is therefore constructed with a jacket to which brine, cooled by means of refrigeration, must be supplied at a temperature of approximately 10 to 15 F. Due to the limitations of effective surface and heat transmission per unit thereof, this direct cooling is sometimes augmented by the introduction of conditioned air which is blown directly upon the surface of the material.

No conditioning is required for the room in which this process is carried on other than to maintain an absolute humidity which will afford freedom from condensation on the cooled material and upon certain cool surfaces which cannot be effectively insulated. General practice is that of applying summer air conditioning—maintaining the room at a dry bulb of approximately 65 F with a dew point of approximately 50 F. A heating function can easily be added at little extra cost. Air that is to be directly introduced in the shredder, however, must not be heated at any time.

While fairly close regulation of temperature is desirable, the dew point may be permitted to vary considerably provided it does not rise sufficiently to cause condensation on the exposed material. The spray type dehumidifier is well adapted and has the advantage, over cooling coils, of affording a considerable amount of intermediate season cooling without the use of refrigeration.

Step 4—Aging. When shredding is complete, the crumbs are packed in metal containers and placed in the aging room to mature for from 48 to 72 hours. The process is one of equalization and chemical reaction. The heat actually generated in this reaction is very slight, but inasmuch as the quality and viscosity of the ultimate spinning solution are materially affected, uniformity of aging is essential to the quality and efficiency of subsequent processes. Departures beyond certain limits in aging may give a product which actually cannot be xanthated.

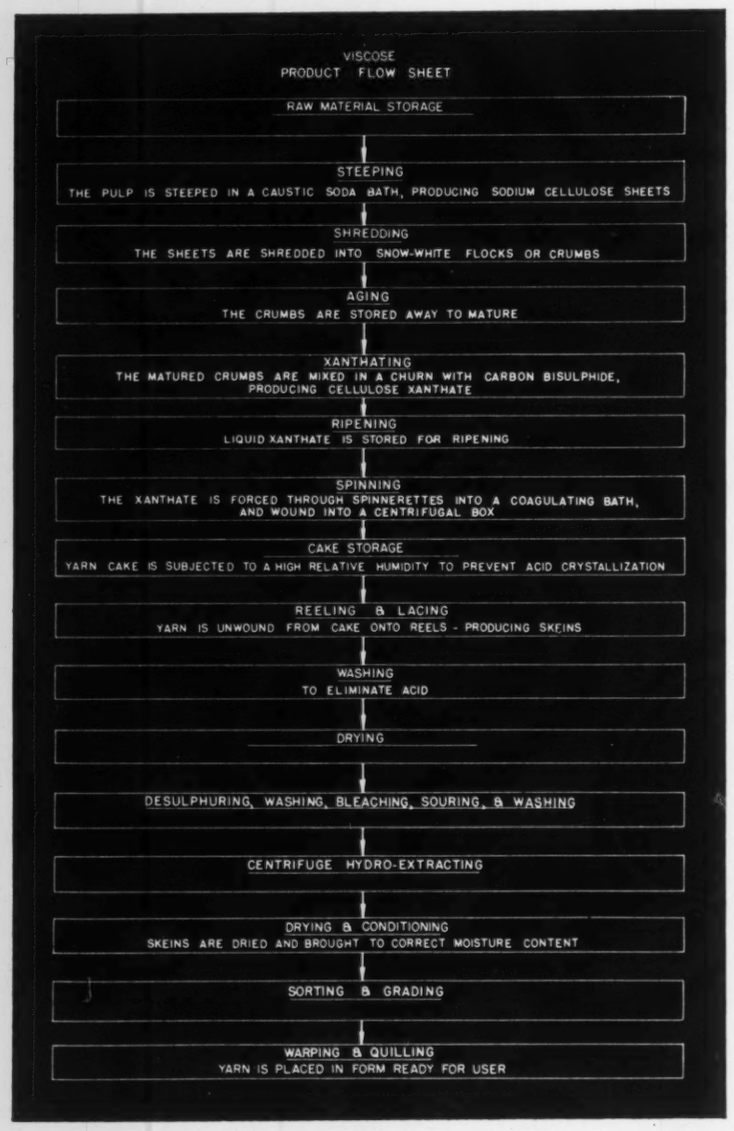
Since the crumbs are stored in metal containers, a system embodying only the elementary function of heat removal with regard to humidity is applicable. This means the circulation of brine cooled by refrigeration through pipe coils suspended from the ceiling. The aging room should be well insulated. Practically no heat is generated within it, so a very uniform temperature with a very moderate air movement is readily obtained. The necessity of placing severe restrictions upon air movement may be attributed to the fact that the rate of heat transfer from the surface of the container to air is a function of air movement or velocity. Hence, with non-uniformity of air movement, the mean effective temperature differences and the temperature of the material within the container will readjust according to air movement.

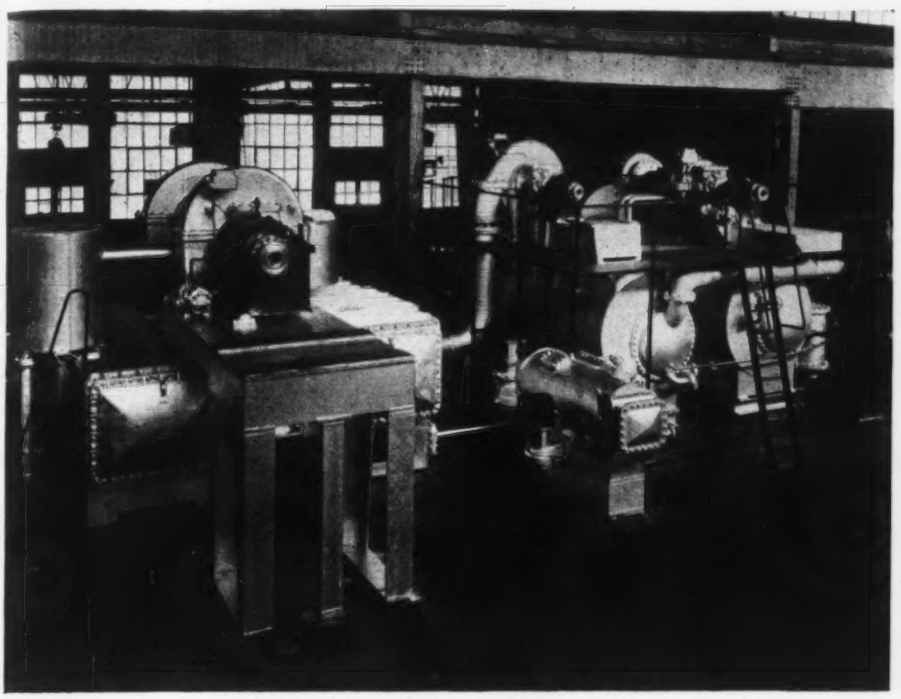
The heart of the problem of regulation with brine coils is a fine regulation of initial brine temperature. The turbine driven centrifugal refrigerating machine with automatic turbine speed variation has met this condition most satisfactorily.

Step 5—Xanthating or Churning. Upon completion of the aging process, the matured crumbs are transferred to a churn where a measured quantity of carbon bisulphide is added.

The churn, which is in the form of a horizontal cylinder, is then rotated about a horizontal axis through its ends. This mixing causes the carbon bisulphide to combine with the alkali cellulose crumbs to form the orange gelatinous mass known as cellulose xanthate.

If any chemical reaction takes place in the previous process, it is most certainly of


VISCOSE
PRODUCT FLOW SHEET
RAW MATERIAL STORAGE
STEEPING
THE PULP IS STEEPED IN A CAUSTIC SODA BATH, PRODUCING SODIUM CELLULOSE SHEETS
SHREDDING
THE SHEETS ARE SHREDDED INTO SNOW-WHITE FLOCKS OR CRUMBS
AGING
THE CRUMBS ARE STORED AWAY TO MATURE
XANTHATING
THE MATURED CRUMBS ARE MIXED IN A CHURN WITH CARBON BISULPHIDE, PRODUCING CELLULOSE XANTHATE
RIPENING
LIQUID XANTHATE IS STORED FOR RIPENING
SPINNING
THE XANTHATE IS FORCED THROUGH SPINNERETTES INTO A COAGULATING BATH, AND WOUND INTO A CENTRIFUGAL BOX
CAKE STORAGE
YARN CAKE IS SUBJECTED TO A HIGH RELATIVE HUMIDITY TO PREVENT ACID CRYSTALLIZATION
REELING & LACING
YARN IS UNWOUND FROM CAKE ONTO REELS - PRODUCING SKEINS
WASHING
TO ELIMINATE ACID
DRYING
DESULPHURING, WASHING, BLEACHING, SOURING, & WASHING
CENTRIFUGE HYDRO-EXTRACTING
DRYING & CONDITIONING
SKEINS ARE DRIED AND BROUGHT TO CORRECT MOISTURE CONTENT
SORTING & GRADING
WARPING & QUILLING
YARN IS PLACED IN FORM READY FOR USER



Centrifugal refrigerating machines cooling brine at a rayon plant

minor magnitude, but in this process the reaction becomes more or less violent. A very considerable amount of heat is generated and explosive fumes are evolved. These fumes are likewise toxic when present in a greater proportion than one part to 20,000.

Inasmuch as it is essential to stop this reaction within close predetermined limits, its rate must be kept under control throughout the process. This may be accomplished by circulating refrigerated brine through the jacket of the churn or by blowing dehumidified air on the churn. When the reaction is complete, the churn is flushed out with air, but consideration must be given to the tolerances of operation and provision made for diluting and removing the explosive and toxic fumes.

The requirement for that and the requirements for eliminating variations in room temperature create the need for conditioning the churn room, which is normally carried at a temperature of 75-80 F summer and winter. Inasmuch as the material in process is completely enclosed, humidity does not matter.

In the practice of conditioning churn rooms, it has long been recognized that maximum freedom from the explosion hazard requires continuous introduction of a relatively large volume of outside air. Practice has established an air change of approximately 20 times per hour. The consequences of an explosion, however, are such that leaning in the direction of safety is a wise and sensible procedure.

Naturally, all of the air supplied to the churn must be taken from out-of-doors, conditioned and then wasted. The needs of cleanliness, of a close automatic regulation, and of economical intermediate season operation point toward the system in which the air is treated with a spray of water.

Step 6—Ripening. From the xanthating process, the solution is passed through the "vissolver" in which sodium hydroxide is introduced. The heat evolved in the ensuing reaction is absorbed by means of a jacket which requires refrigerated brine. The solution is then filtered thoroughly and stored in the ripening tanks in which it is also deaerated.

A mild chemical reaction continues as a result of which a very slight amount of heat is generated. The fixing of viscosity within narrow limits is vital to the spinning process which follows, and since, among other factors, time and temperature affect viscosity, it is of vital importance accurately to control the temperature of the ripening room. Inasmuch as the solution is in tanks, humidity is of no consequence, except as it may affect the economics of the design of the air conditioning system.

Refrigeration of the utmost in reliability is essential, for if the solution rises above a certain critical temperature, coagulation will destroy its value as a spinning solution.

Suitable systems are the same as those indicated for the aging room. Uniformity of temperature is of equally great importance, because a variation of one or two degrees in the temperature of the solution may add or subtract as much as eight or ten hours to or from ripening time.

Step 7—Spinning. When the viscose solution is ready for conversion into filaments, it is fed to the spinning machine under pressure, where it is pumped through platinum spinnerettes or jets into a coagulating bath. Each spinnerette is perforated by a number of holes corresponding to the number of filaments which are to make up the ultimate thread.

The number of filaments per thread of yarn range normally from 100 to 500, although ropes of 1500 are produced.

The coagulating solution consists generally of dilute sulphuric acid and sodium sulphate, but sometimes ammonium sulphate and glucose are also added. As the viscose or spinning solution meets the acid solution, it is quickly coagulated and the jets of liquid viscose are converted into fine filaments of regenerated cellulose. After leaving the coagulating solution, the thread passes under a guiding hook, up and over a revolving feed wheel, down through a vertical reciprocating tube, and into a suitable collector.

Two forms of collectors are in use—the bobbin and the centrifugal or "topham box." The centrifugal box is more generally used. This box rotates rapidly on a vertical spindle. The centrifugal force throws the thread to the side of the box, where it builds up into an annular cake, the required amount of twist being imparted by the rotation of the box.

The purposes for which air conditioning are needed in this process are manifold. A new chemical reaction is set up, the equilibrium and rate of which are materially affected by temperature and humidity. Acid fumes are released and employees must be thoroughly and completely protected from them. These fumes are highly irritating to the membranes of the eye and, entirely aside from human considerations, can readily have a grave effect upon the economics of production.

The acid fumes must be collected, and the practical manner of doing this is to place the chamber in which the reaction takes place under a negative pressure, thus introducing a rapid flow of air from the spinning room to this chamber and thence outward through the exhaust system. Room conditions are important only in the sense that fluctuations there mean fluctuations in the air passing through the reaction chamber. All the air delivered to the spinning room should be exhausted to outside at a point of sufficient elevation to afford perfectly safe diffusion.

Year 'Round Air Conditioning Essential

Year 'round air conditioning is essential with great emphasis on the heat requirements of the winter air conditioning function. Spray equipment is essential, since room temperatures not much in excess of from 80 to 85 F are desirable and can be had with evaporative cooling during all but a few exceedingly warm summer days. Relative humidity depends somewhat upon the spinning process, which will require a minimum of 70 per cent. A tremendous amount of heat is required to bring winter air up to this terminal condition.

Slight variations from these conditions result in the formation of certain crystals inside the machine and upon the thread, which is productive of trouble.

The volume of air has been determined by practice to vary between 35 and 60 cfm per spindle, the limits of variation apparently being determined by the *denier* (weight per unit length) which, in a sense, is a measure of the tonnage of production and, in that same sense, is a measure of the pounds of reagent entering into the chemical reaction.

The author does not know whether volume is actually determined by the requirements of the product for air or by the limitations of the spinning frame housing. Acid fumes must be kept within that housing and access to enclosed parts is essential. The requirements of the product for air are likewise unknown but, if "horse sense" may be called upon, it seems to say that present practice calls for a very large volume of expensive air. This seems to point toward an improved housing which should repay its cost by materially reducing the cost of air conditioning.

Although considered separately, the problem of conditioning the spinning room involves that of conditioning the reeling room, inasmuch as economies can be effected by feeding the spinning room exhaust with air that is permitted to flow from the conditioned reeling room.

[To be concluded]

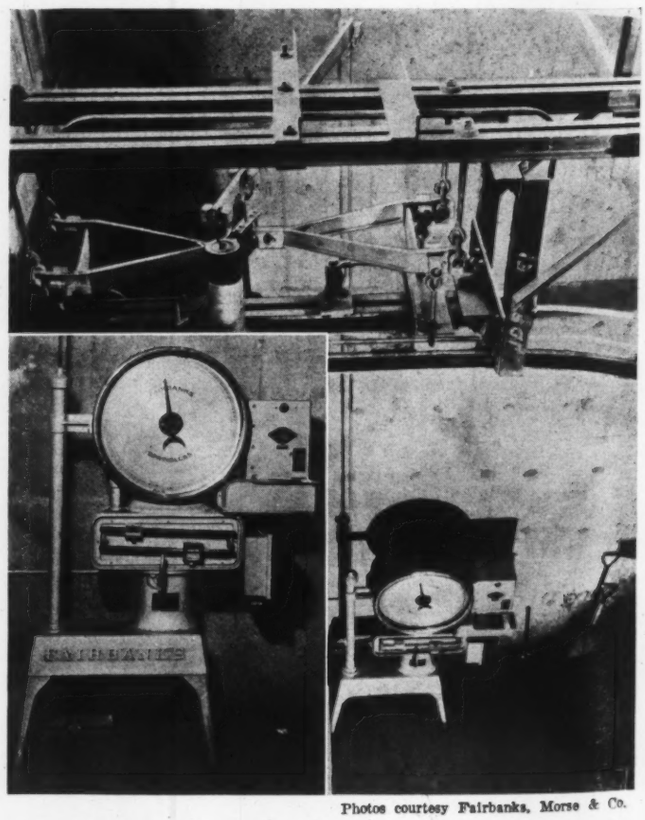
❖ ❖ ❖

Automatic Weigher Records Coal for Heating School

A PRINTED record of every pound of coal used for heating the new Southwestern high school, St. Louis, will be provided by the automatic recording weigher illustrated.

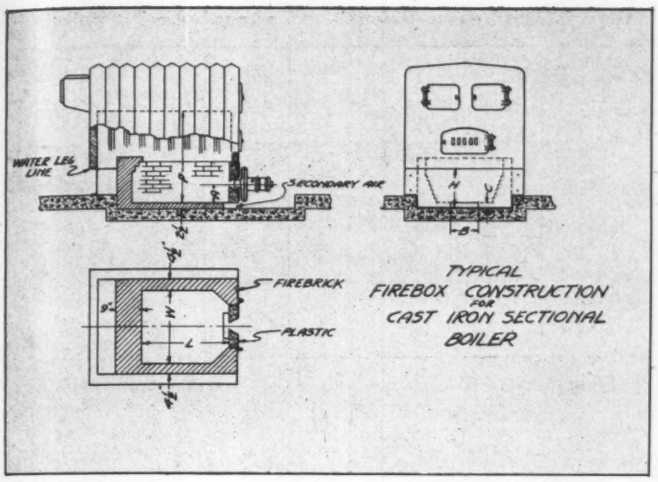
The weighing equipment consists of an overhead monorail track scale, an indicating dial, and the automatic weigher. Coal is unloaded into a receiving pit from which it is carried by a conveyor bucket into the furnace hoppers. A time switch is arranged so that each load carried along the conveyor rail in the suspended bucket is automatically stopped and weighed, the weight being recorded on a tape. Empty buckets returning over the rail are not stopped or weighed. The average load of coal is about 1000 lb.

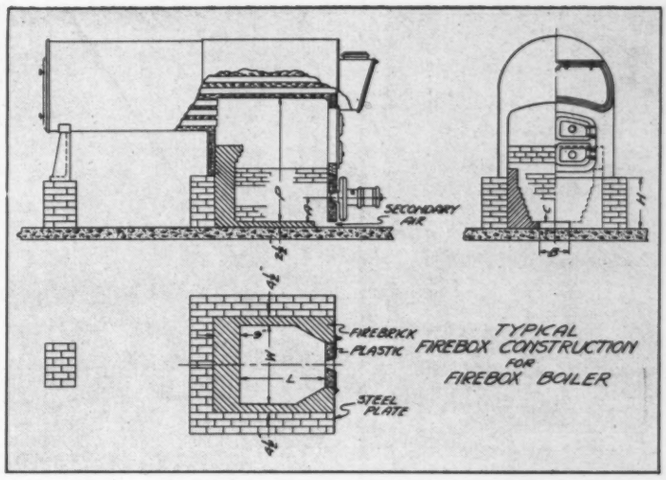
The building will have 45 class rooms and will accommodate approximately 1750 students.



Photos courtesy Fairbanks, Morse & Co.

View (from above) shows overhead monorail track scale and automatic weigher. Insert shows close-up of dial and weigher





Combustion Chambers for Oil Fired Boilers

By Kalman Steiner*

NUMEROUS heating plants that have been in use during the past winter will require some sort of attention before being placed back in service next fall. Among the necessary maintenance operations in many cases will be the repair or rebuilding of boiler fireboxes, including those of oil fired installations.

Usually, the plant engineer will have observed some features of his plant performance during the heating season which will lead him to believe that some modification should be made in the firebox design for his particular installation, these changes to be incorporated into the new masonry. In many cases his observations, if properly interpreted and in turn correctly embodied in firebox alterations, will result in significant improvements in boiler operation with consequent increases of combustion efficiency. In fact, were it possible to collect a large group of such operating observations, facts, and data, and then analyze this material from the viewpoint of operating performance, it would almost certainly result in the establishment of some very concrete rules for combustion chamber design. Data of that nature would be as valuable as test data secured in laboratories (and but little laboratory work has been performed on this subject), and even more valuable than the figures secured by tests made in the field, for such tests are usually carried on under ideal conditions. Just what data the operating man should attempt to collect will become apparent as this discussion proceeds.

Before the theoretical importance of operating data can be explained, it is necessary to review the present status of available information on the subject, and to discuss the principles now used in combustion chamber design and construction. Much has been said and written on this matter of fireboxes for oil burners, and no doubt a great many differences of opinion prevail. Nevertheless, a few simple rules can be set up that will cover—to a large extent—what is considered to be good practice. It must be understood that these rules are intended only to indicate tendencies, for they are based on averages—and averages are, in a sense, capable of being misused, unless allowance be made for possible individual deviations from them. Thus, if data were based upon figures from a hundred similar boiler installations, and the rate of heat release per cubic foot of combustion volume varied from 26,000 to 40,000 Btu per hr, with the average at 33,000, then the lowest and highest would each be about 21 per cent from the average. Should an installation then be constructed upon the average, whereas it actually requires the lower value, serious operating difficulties might result. Hence, the reservation must be made that the figures used in this article are only averages. When more and better data are available, it may be possible to make more definite statements concerning design methods.

The firebox was largely intended, in early installations, merely to serve as a refractory chamber within

Among the maintenance operations necessary in many cases to put heating plants in condition for service next winter will be the repair and rebuilding of boiler fireboxes, including those of oil fired installations. Combustion space, shape of the combustion chamber, secondary air port area and other details are discussed here, and the author concludes with an appeal to men in the field to submit data on their experiences in order that more or less theoretical rules may be proved—or disproved. Tables of dimensions for combustion chambers for boilers are given

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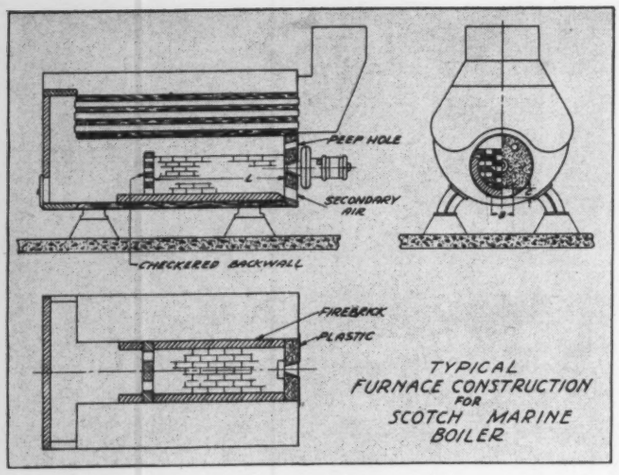


Combustion Chamber Dimensions for Scotch Marine Boilers

W Diameter of Firebox, In.	Boiler Horse-Power	Diameter of Shell, In.	Length of Tubes, In.	A Nozzle to Floor, In.	B x C Area of Secondary Air Port, Sq In.	L Length of Firebox, In.
26	20	44	78	10	28	62
26	25	44	96	11	35	78
26	30	48	96	11	42	90
26	35	48	114	12	49	108
30	40	54	96	12	56	90
32	50	60	114	13	70	102
36	60	72	114	14	84	98
36	75	72	144	15	105	122
42	100	84	160	16	140	120
45	125	90	160	17.5	175	130
45	150	90	174	18	210	155
48	175	96	174	20	245	160
52	200	102	174	21	280	155

Note: If the total secondary air port area cannot conveniently be placed below the burner, an auxiliary air port may be placed above burner. Sometimes refractory is used only for the front, and the side-walls are left bare; this practice is successful on some types of burners, but generally the catalytic effect of glowing firebrick is needed around the fire.

which combustion could occur, the refractory walls and floor serving also to protect the boiler setting from injury from the fire. It was soon learned that when the refractory attained the temperature of incandescence the presence of these white-hot surfaces exerted an influence over combustion, causing it to proceed more rapidly and more completely, and thereby yield a cleaner fire with less excess air needed. This is one of those curious behaviors of substances which chemists describe as a *catalytic effect,* meaning that a substance exerts an influence upon a chemical reaction by its very presence and without seemingly participating itself in the process. This is a point to be borne in mind when considering fireboxes for Scotch marine boilers. Finally, it also became apparent that heat was radiated from the hot brick surfaces directly up into the boiler shell, and that therefore the form of the refractory had to be considered from the viewpoint of increasing the exposure of boiler surface to the heat radiated from the brickwork.

Rules for Firebox Construction

The first rule can now be stated: An oil burner should be mounted in such position, and the firebox should be constructed in such manner, that the combustion can occur within a space which, so far as possible, is lined with refractory, and this refractory should be so formed and arranged that it can become incandescent from the fire.

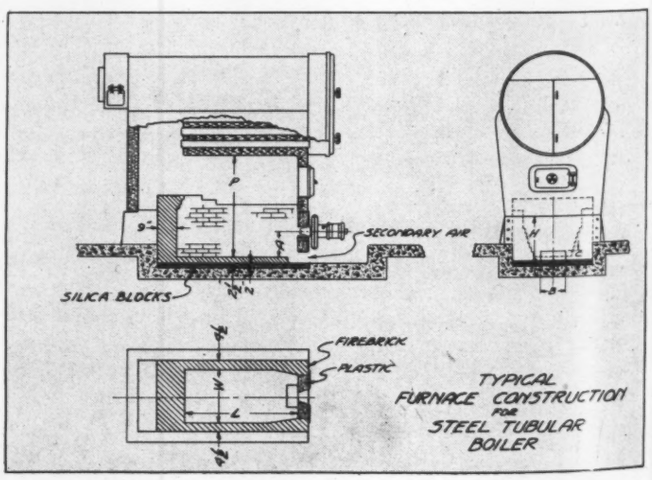
It is rarely (if ever) possible to have the combustion space entirely lined with refractory. Just what percentage of such lining can usually be arranged for, and what minimum can be considered as acceptable for various types of boilers, will appear from the data developed below. For our purposes here, it is necessary to define combustion space as the total free volume in which combustion can occur and in which the gases of combustion can expand and undergo turbulence before entering flue passages or striking against water tubes, or passing beyond the bridge wall.

From this rule it follows that the usual setting provided on a low pressure hand fired boiler is not sufficient for oil firing. It is necessary either to raise the boiler on a high base, or to pit the floor under the boiler firebox. Raising of a boiler is seldom a feasible procedure on existing installations, although frequently new boilers are so set. But in most conversion installations, as well as in a great many new boiler installations, the floor under the boiler is pitted to a depth that will permit of proper burner setting and proper firebox construction. It will be seen, below, that pitting (or raising) the boiler serves at least two other purposes as well. But our first rule requires that as much of the combustion chamber as possible be surrounded by hot firebrick.

Of course, were the presence of firebrick around the oil fire the only requirement, then this percentage could easily be increased by merely extending upward the walls of any firebox. But such practice would cut off valuable heating surface from exposure to the fire or to hot gas—mostly from direct exposure to the fire. And since the rate of heat transfer into the boiler is always much higher for surface exposed to the fire or to glowing firebrick than for surface which merely contacts hot gases, it is seen that a serious reduction of boiler capacity might result from undue extension upward of the firebox walls. Usually, the side walls of a firebox are carried up only to a point just above the rivet seam.

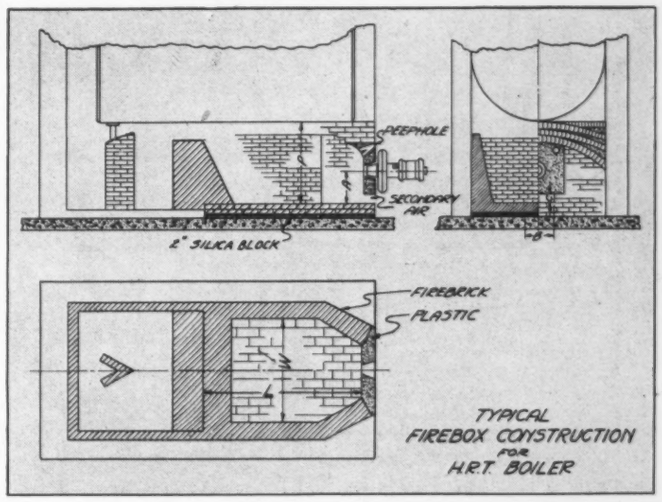
Shape of Combustion Chamber

The second rule pertains to the shape of the combustion chamber. Actually, the shape of the box is not as important as might be indicated by the amount of



Combustion Chamber Dimensions for Horizontal Return Tubular (H. R. T.) Boilers

W Width of Firebox and Diameter of Boiler, In.	Boiler Horse Power	A Nozzle to Floor, In.	B Width Secondary Air Port, In.	C Height Secondary Air-Port, In.	P Height Combustion Chamber, In.	Length of Boiler, Ft	L Length of Firebox, In.
60	80	14.0	25	4.5	50	16	67
60	90	15.0	28	4.5	50	18	83
66	100	16.0	24	6.0	54	16	77
66	110	16.5	26	6.0	54	18	85
72	125	17.5	30	6.0	58	16	83
72	150	18.0	35	6.0	60	18	96
72	165	19.0	39	6.0	66	20	96
78	180	20.0	42	6.0	72	18	98
78	200	21.0	47	6.0	78	20	90



discussion which has transpired in regard to it. Reduced to simplest terms, our second rule can be stated as: Let the shape of the refractory conform as closely as possible to the shape of the fire, but avoid having the refractory surface so close to the fire that flame impingement will cause carbon deposit.

As a rule, the only point of contact of fire on brickwork should be the bridgewall. The side walls and the arch on front should be out of range of the flame or path of fuel oil. Since the usual oil fire broadens out from a point just in front of the nozzle, it follows from the rule that the front must be shaped in somewhat of a hollow cone, with the nozzle at the apex. Obviously, the upper part of such a cone cannot be carried very far, and only the bottom and walls carry the refractory any considerable distance beyond the front. In many cases the archwork over the fire is built outward so as to complete the encircling of the fire for as great a distance as possible. But such archwork must be built with great care, so that sufficient clearance is maintained from the fire; otherwise oil will strike the arch and form carbon. The same caution should be observed in curving or bending the side walls inward as they approach the front. Many boiler masons and oil burner men, in their attempts to bring the refractory up close to the fire, do not leave enough clearance, and carbon inevitably is deposited. The floor must also be at an adequate distance from the fire, and for the same reason. The sidewalls are generally sloped inward, for the double purpose of keeping the refractory shape in conformance to the fire shape, and of creating an angle to the surface which will reflect heat upward into the boiler shell.

One other factor must be considered in respect to the shape of the firebox, and that is the matter of protecting from the fire those portions of the boiler and setting which can be injured by heat. Included in this category are rivet seams, mud legs, blow-down pipes and flanges, grate brackets and concrete foundations. The typical fireboxes illustrated show how this protection is accomplished. In certain cases where unusual conditions prevail, it is necessary to employ special insulating materials, such as silica blocks applied against concrete that has been waterproofed, or where it is desired to minimize the amount of heat lost from the firebox by conduction through the concrete setting.

A Controversial Point—Combustion Volume

The third rule governs one of the most important aspects of the entire subject of combustion chamber design: Combustion volume.

Here we encounter controversial opinion; just a little reflection will reveal that the differences exist mainly because of the chaotic state of knowledge on the subject. In the first place, various sorts of methods are used for expressing the factor of combustion volume; some writers express data in terms of cubic feet of combustion volume per boiler horsepower, others reduce it to Btu release per cubic foot or gallons of oil burned per cubic foot; again an expression will be found involving grate area of an equivalent coal fire, etc. This means that some design is based on heat input, while again boiler output is sometimes made the basis. Hence, the factor of efficiency enters, and that is a variable quantity. In the second place, all authors are not in agreement as to what constitutes combustion space. Even the definition for combustion volume given above is ambiguous for certain types of boilers. Thirdly, it is very difficult for a close or accurate statement to be made for most heating plant operation concerning either the load condition or the combustion or firing rate. Fourthly, very little laboratory work has been performed on the subject, and little authentic test data have been published for oil fired heating boilers. No wonder that opinions vary!

Certain arbitrary conditions will have to be set up in this article in

Combustion Chamber Dimensions for Cast Iron Sectional (C. I.), Steel Firebox (S. F.), and Welded Tubular Boilers (W. T.)

Rating Sq Ft Steam Radiation	A Nozzle to Floor, In.	B Width Secondary Air Port, In.	C Height Secondary Air Port, In.	H Height Boiler Setting, In.	P Height Comb. Chamber, In.			W Width of Firebox, In.			L Length of Firebox, In.		
					C. I.	S. F.	W. T.	C. I.	S. F.	W. T.	C. I.	S. F.	W. T.
1,500	7.0	4.5	3.5	14	35	52	35	23	21	20	31	22	34
2,000	8.0	4.5	4.5	14	35	52	35	23	21	20	41	30	40
2,500	8.0	5.5	4.5	16	35	52	35	23	21	21	54	35	57
3,100	8.5	7.5	4.5	18	38	59	39	23	27	21	54	35	62
4,000	9.0	9.0	4.5	18	40	59	41	23	27	27	71	41	61
5,000	9.5	11.5	4.5	18	41	62	39	35	33	27	49	41	83
7,200	10.5	15.5	4.5	20	43	64	45	39	39	32	82	53	84
10,000	11.0	22.0	4.5	20	44	69	50	68	39	39	50	49	65
12,500	11.5	27.0	4.5	20	.....	69	50	.....	44	44	.....	55	73
15,000	12.0	33.0	4.5	22	.....	74	50	.....	50	49	.....	55	80
20,000	13.0	34.0	6.0	25	.....	80	55	.....	54	55	.....	72	90
25,000	14.0	42.0	6.0	30	.....	85	64	.....	56	60	.....	75	90
30,000	16.0	50.0	6.0	30	.....	85	66	.....	62	69	.....	84	92

(See sketches for dimensions)

Information desired of anyone who has had occasion to design or construct an oil burner combustion chamber for a heating plant or allied type of boiler:—

1. **Name, catalogue number and type of boiler.**
2. **Name, or type of oil burner.**
3. **What difficulties, if any, were encountered in operation prior to alterations to firebox? Smoke, carbon deposit, slow or sluggish steaming, refractory trouble, excessive oil consumption, etc.**
4. **What improvements in operation were effected by alteration of design or construction?**
5. **Describe the firebox design, following the method of classification used in this article: burner position, firebox shape, combustion volume, secondary air port area, etc.**
6. **In discussing load carried, be sure to include radiation or heating load and hot water heating load, as well as any special load conditions.**

Mr. Steiner stands ready to classify and interpret operating data and information on design and construction of boiler combustion chambers, appeals to operating men for help. An advance copy of the result will be sent to all who cooperate. Send data to The Editor, HEATING, PIPING AND AIR CONDITIONING, 6 N. Michigan Ave., Chicago, Ill.

order that quantitative statements can be made. It really is not important whether the conditions imposed seem reasonable or not, for they merely define the meaning of the quantities used. Anyone caring to set up other conditions has merely to modify the quantities accordingly. These conditions are: For all steel heating boilers, *S. H. B. I.* ratings are used; for cast iron boilers, guaranteed output ratings of the manufacturer are used; overall efficiency of burner and boiler assumed to be 70 per cent; combustion volume as defined before in this article.

At the risk of being challenged, the author makes the generalization that for boilers rated to carry under 7000 or 8000 sq ft EDR, the average percentage range of combustion volume enclosed within refractory will be 28 to 35, and that for larger boilers the percentage will range between 35 and 45. This statement applies to the three most commonly used types of heating boilers, cast iron sectional, steel firebox and steel tubular. The H. R. T. and Scotch marine are obviously not subject to such generalization, the former because the walls are entirely masonry, and the latter because the firebox is entirely enclosed within water-backed heating surface. But for the first three boilers named, our third rule will be stated thusly: For boilers under 8000 sq ft EDR output rating, design for a heat release of 30,000 Btu per cu ft of combustion volume; for larger boilers, design for a heat release of 40,000. For H. R. T. boilers, the rule is: Design the firebox with a heat release of 30,000 for 100 per cent rating, 37,500 at 125 per cent rating, and 45,000 at 150 per cent rating. For Scotch marine boilers, figure 50,000 Btu release at 100 per cent rating, 62,500 at 125 per cent, and 65,000 at higher ratings.

It will usually be found that when H. R. T. boilers are to be developed beyond 150 per cent no bridgewall can be used, and that when Scotch marines are loaded beyond 100 per cent an extension of the furnace shell will be needed (frequently referred to as a "dutch oven").

Secondary Air Port Area

A great deal more could be said about combustion volume, but it is necessary to proceed to our fourth rule. However, as will be brought out in just a moment, the author hopes that his readers will have a lot to say, not only about combustion volume, but about all the factors determining or related to firebox design or construction.

This rule relates to secondary air port area, and here again qualifying conditions must be described. By far the greater majority of oil burning plants are natural draft, and the most commonly encountered ratio of primary air to secondary air is about one to six, or in other words, about 85 per cent of the air required for combustion is induced as secondary air by natural draft. And with the increasingly general use of draft control apparatus of one sort or another, overfire draft will be found to average between 0.05 and 0.15 in. water. For these conditions, then, secondary air port area should be 1 sq in. per 100 sq ft EDR load, or 1.4 sq in. per developed hp. This is our rule.

An Appeal for Help

These rules represent what today can be called theoretical design. But how closely do these more-or-less theoretical rules coincide with actual results? Who can sustain them by proved performance? Or who can disprove them or improve upon them, by proved performance results? That is what should be known, and the author believes that operating men have the answer. The next thing is to gather data which operating men have accumulated in their experience. The author holds himself ready to sort out, classify, analyze, and interpret the material and to publish the results, if operating men will turn it in. On this page there appears a summary of the information desired. It is earnestly requested that anyone having had experience of any nature in oil burner firebox design and construction send in to the editor his opinions, data or ideas. Especially it is hoped that performance results tied up with design methods can be described by the men in the field who are responsible for the operation of heating plants.

Building Managers Must Know Air Conditioning

Air conditioning is rapidly becoming a major issue in building management, an editorial in a recent issue of *Buildings and Building Management* points out, and the subject is still controversial. No one responsible for present and future earnings of office and apartment buildings is justified in permitting any personal opposition to air conditioning to blind him to the possibility that he may be mistaken, advises the editorial. Even though the manager may be personally convinced that his arguments against air conditioning are stronger than are those in its favor, his responsibilities as the chief executive of his properties obligate him to hedge his decision to protect himself against a serious error in judgment.

Executives who are not prepared to deal with air conditioning when it confronts them are gambling needlessly, for if they are wrong and delay their decisions when air conditioning is necessary, the losses may be serious, the editorial warns.

How to Avoid Trouble on Cooling Jobs With Evaporators Above Condensers

[Part 3—Preventing Liquid Refrigerant from Vaporizing]

By William Goodman*

Where it is necessary to place the evaporating coils some distance above the condenser or receiver—as it is in many air conditioning installations—problems are created which must be recognized and properly dealt with if trouble is to be avoided. The author describes this month methods for preventing liquid refrigerant from flashing to vapor in vertical pipe lines, to prevent reducing capacity of expansion valves.

WHERE it is necessary in an air conditioning or refrigerating installation to place the evaporating coils some distance above the condenser or receiver, problems are created which must be recognized and properly dealt with if subsequent difficulties are to be avoided.

Among the principal difficulties are those due to the incorrect selection of expansion valves. Valves must be chosen not only for the smaller pressure difference, due to the lower pressure existing at the top of vertical pipe lines, but their selection must also allow for the small amount of vapor formed in the vertical line. Although the actual weight of vapor is small, the volume occupied by it is relatively large—so much so that it has a substantial effect upon the capacity of an expansion valve.

Part 1, published in April, discussed the pressure variation in a vertical "Freon" pipe line, how to compute the pressure at any point, and how to insulate the piping. In Part 2, which appeared in the May issue, the selection of "Freon" expansion valves was explained, and it was pointed out that valve capacity depends upon the pressure difference across the valve, and the density of the refrigerant. It was shown how refrigerant vaporizing in the liquid line reduces the rated valve capacity.

Methods of Preventing Liquid from Flashing

There are several methods that can be used to reduce the amount of vapor reaching the inlet of an expansion valve. These methods fall into two broad classes, depending upon whether some of the liquid is actually prevented from flashing in the vertical line, or whether the vapor which does form is condensed ahead of the expansion valve.

Subcooling—Of the methods that can be used to prevent liquid from flashing in the vertical line, one of the most obvious is to subcool it to as low a temperature as is possible with the available condenser water. The lowest temperature to which the liquid can be subcooled is, of course, some temperature slightly above that of the water entering the condenser. In order to accomplish this subcooling, the liquid must be completely separated from the vapor. As long as vapor is mixed with the liquid, subcooling of the liquid cannot take place; the vapor will condense and the water will pick up the latent heat of the condensate rather than the sensible heat from the subcooling of the liquid.

For effective subcooling, a subcooler separate from the condenser should be provided, as illustrated in Fig. 4. Unless the vertical rise is small, this method will not prevent entirely the flashing of liquid in the vertical line. However, if the initial temperature of the condensing water is not too high, it can be used to reduce substantially the amount of liquid which will flash in the vertical liquid line.

Subcooling the liquid, as in Fig. 4, will cause the pressure in the vertical line to fall faster than if the liquid were not subcooled, resulting in a smaller pressure difference across the valve. On the other hand, the density of the mixture at the inlet of the valve will be increased because of the subcooling. Whether the increase in density will be large enough to more than offset the decrease in pressure difference is a question that can only be answered definitely for a particular set of circumstances. However, where the vertical rise is moderate, the increase in density thus obtained will usually more than offset the decrease in valve capacity due to the smaller pressure difference. Another very important result of subcooling the liquid "Freon" with the condenser water is that the refrigerating capacity of the system is increased thereby.

Use of a Pump—A second method, which would prevent any flashing whatever in the vertical liquid line, would require the use of a pump. The liquid "Freon" would be pumped from the liquid receiver up to the expansion valve. In this way, the pressure in the liquid line could be increased to such a point that all possibility of liquid vaporizing in the vertical pipe line would be eliminated. While this method is entirely feasible theoretically and has considerable promise, there are, unfortunately, no pumps available with packing suitable for high pressure "Freon." Although a small amount of leakage is permissible through the packing of water pumps, the packing of a "Freon" pump must be absolutely tight if refrigerant losses are to be avoided. For this reason, this method cannot be considered among those available at present, although—as mentioned—it has possibilities.

Eliminating Vertical Liquid Line—The vertical liquid

*The Trane Co. Member of Board of Consulting and Contributing Editors. Copyright, 1937, by William Goodman.

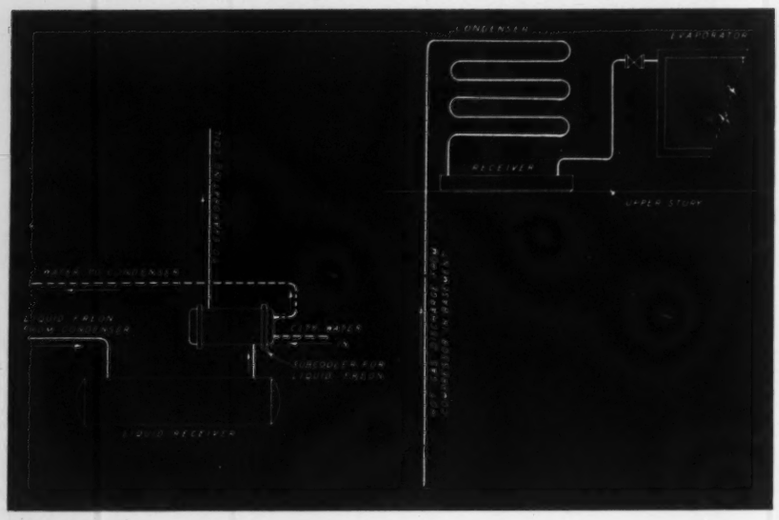


Fig. 4 (*left*)—Method of connecting liquid subcooler into the liquid line for subcooling the liquid refrigerant *before* it enters the vertical liquid line. Fig. 5 (*right*)—Method of placing the condenser and receiver alongside of an elevated evaporating coil to prevent liquid flashing in the vertical line

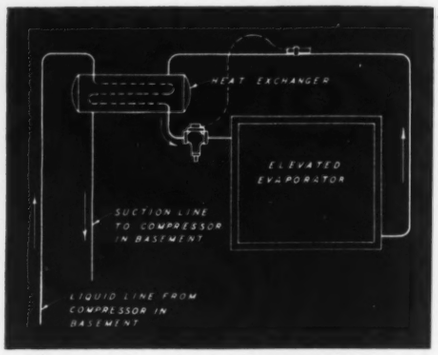


Fig. 6—Method sometimes used for subcooling the liquid by means of the suction vapor leaving the evaporator. The velocity of the vapor in the small length of suction pipe which rises vertically up to the heat exchanger should be kept above 1000 fpm in order to carry the entrained oil along

line can be eliminated entirely by placing the condenser alongside the elevated evaporating coil, as illustrated in Fig. 5. The hot gas is then piped from the compressor to the elevated condenser. Where city water is used and water pressures are low, this method usually requires pumping the water up to the condenser, thus adding an additional piece of equipment and increasing the operating cost. However, where a roof cooling tower is used, or where city pressure is sufficient to raise the water to the condenser, this method is one of the best. Provision should be made for returning oil, which separates from the vapor, to the crankcase.

Use of Heat Exchanger—Where flashing of the liquid does take place in a vertical liquid line, it is possible to condense the vapor ahead of the expansion valve by means of a heat exchanger, as illustrated in Fig. 6. The cold vapor leaving the evaporator is used to condense the flash vapor mixed with the liquid approaching the expansion valve. Because the specific heat of superheated "Freon" vapor is low (only 0.15), the weight of vapor which can be condensed with it is limited. If the vapor is superheated as much as 40 F, which is improbable, it can pick up only approximately 6 Btu per lb (40×0.15). Since the latent heat of "Freon" is about 60 Btu per lb, the flash vapor in the liquid line can be entirely condensed only if its weight is less than about 10 per cent of the total quantity of "Freon" flowing through the expansion valve. In other words, if 15 per cent of the "Freon" in a liquid line is vapor, less than 10 per cent will be condensed if this method is used; the balance will flow with the liquid to the expansion valve. Fig. 8 may be used to find the percentage of vapor in a mixture. In a short vertical line, where little vapor is formed, this method can probably be applied successfully. In such cases the weight of vapor may be so small that the liquid may even be subcooled by the cold vapor after the flash vapor has been completely condensed.

The use of a heat exchanger when the evaporating coil is at some vertical distance above the condenser results in some increase in the capacity of the valve due to the reduction in the weight of vapor approaching the valve and to the possible subcooling of the liquid. However, it should be clearly understood that the installation of a heat exchanger such as the one illustrated in Fig. 6 has no effect whatever upon the refrigerating capacity of the system. Even though the amount of flash gas entering the evaporator coil is reduced, the volume of each pound of vapor in the suction line has been increased due to the superheating. Because of the increase in the volume of vapor entering the compressor, the capacity of the compressor will be reduced. As a rule, this reduction will just about offset the increase in the refrigerating capacity due to the condensation of the vapor and possible subcooling of the liquid. For small jobs it would seem desirable, in the interest of economy and simplicity, to install the necessary larger expansion valves rather than a heat exchanger. On larger jobs,

Table 1—Saturation Pressures of "Freon"
(*From A.S.R.E. Data*)

Temperature Deg Fahr	Gage Pressure Lb per Sq In.	Temperature Deg Fahr	Gage Pressure Lb per Sq In.
30	28.5	80	84.1
32	30.1	82	87.0
34	31.7	84	90.1
36	33.4	86	93.2
38	35.2	88	96.4
40	37.0	90	99.6
42	38.8	92	103.0
44	40.7	94	106.3
46	42.7	96	109.8
48	44.7	98	113.3
50	46.7	100	116.9
52	48.8	102	120.6
54	50.9	104	124.3
56	53.1	106	128.1
58	55.4	108	132.1
60	57.7	110	136.0
62	60.1	112	140.1
64	62.5	114	144.2
66	65.0	116	148.4
68	67.5	118	152.7
70	70.1	120	157.1
72	72.8	122	161.5
74	75.5	124	166.1
76	78.3	126	170.7
78	81.2	128	175.4

because of the size and number of expansion valves which might be required, the installation of a heat exchanger may be worthwhile.

Another method, illustrated in Fig. 7, can be used not only to condense entirely the flash vapor formed in the vertical liquid line, but also to subcool the liquid almost to the temperature being maintained in the evaporator. Liquid "Freon" is expanded directly into a heat exchanger placed beside the elevated evaporating coil. The mixture of liquid and flash vapor in the vertical riser flows through this heat exchanger on its way to the expansion valve. As a result, the vapor in the liquid line is entirely condensed and the liquid itself is subcooled almost to the temperature corresponding to the suction pressure.

No Loss of Refrigerating Capacity

There is no loss in refrigerating capacity due to a portion of the liquid "Freon" being used in the heat exchanger. As has been stated, a certain weight of liquid must inevitably flash into vapor as it expands from the pressure in the liquid line to the pressure being maintained in the evaporating coils. This flashing is due to the cooling of the liquid "Freon" while flowing through the expansion valve. No matter how the cooling of the liquid is accomplished, a certain weight of "Freon" must flash into vapor. This being the case, it is immaterial whether this flashing takes place in the evaporator or in the auxiliary heat exchanger. The refrigerating capacity of the system will be unaffected by the use of a heat exchanger such as that shown in Fig. 7.

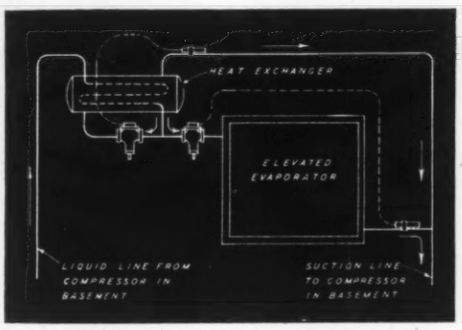


Fig. 7 — Method of subcooling liquid by expanding liquid "Freon" directly into a heat exchanger

The auxiliary heat exchangers of Figs. 6 and 7 should be placed in such a position that the "Freon" leaving them will flow down to the expansion valve at the evaporating coils. This is desirable because the density of the mixture leaving the heat exchanger is greater than that of the mixture entering from the vertical riser. The velocity of the vapor in the small length of suction pipe which rises vertically up to the heat exchanger of Figs. 6 and 7 should be kept above 1,000 fpm in order to carry the entrained oil along.

When using the last two methods described, it is necessary to remember that the pressure ahead of the expansion valve will be exactly the same as if the auxiliary heat exchangers were not used. The only effect of these heat exchangers is to increase the density of the mixture flowing to the expansion valve.

In these articles an attempt has been made to give a clear summary of the action of liquid "Freon" in vertical pipe lines. Difficulties can be prevented if care is exercised in the selection and installation of expansion valves, and if the portion of the liquid line in the unrestricted zone is properly insulated.

Appendix

In preparing Figs. 2 and 3, the percentage of liquid which would flash to vapor when expanded between two known pressures was computed by the well-known method to be found in elementary refrigeration texts. These computations enabled the chart of Fig. 8 to be drawn. This chart gives the percentage of liquid (by weight) in a mixture after the liquid has been expanded from some initial pressure to a lower final pressure. The data on this chart, together with the densities of "Freon" liquid and vapor given by *A.S.R.E.* tables, were then used to compute the density of the mixtures of "Freon" liquid and vapor. These data were then plotted on Fig. 3.

Fig. 2 was prepared by a method of arithmetical integration by the use of the following formula:

$$144\ (\triangle p) = -(\triangle h)\, d$$

where

$\triangle h$ = a small rise in the vertical pipe line, in ft,

d = density of the mixture at any point in the vertical line, in lb per cu ft,

$\triangle p$ = reduction in pressure in the length $\triangle h$, in lb per sq in.

For instance starting with 100 per cent liquid "Freon" at some assumed initial pressure, the pressure reduction due to a vertical rise of one foot was calculated. The reduced density at the new final pressure was then read on Fig. 3. This reduced density was then used in the formula to compute the reduction in pressure for a further rise of one foot. The density at this second final pressure was then read again on Fig. 3 and used to compute the pressure reduction due to another rise of one foot. So—foot by foot—the pressure at various vertical distances was computed for a given initial pressure. In order to plot Fig. 2, it was necessary to repeat this performance several times with different initial pressures.

In the actual computation of Fig. 2, in order to achieve as great a degree of accuracy as possible, a table of densities was used instead of Fig. 3. This table gave the final density for final pressures at intervals of one pound for the several different initial pressures used in plotting the curves of Fig. 2.

[*Concluded*]

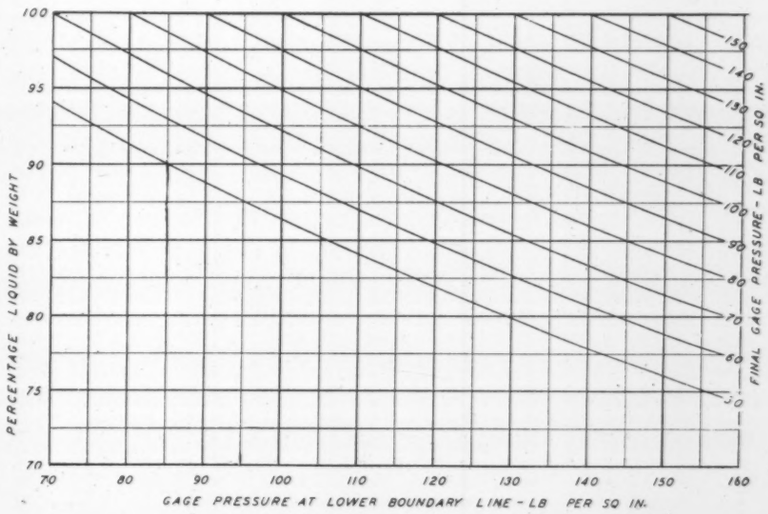
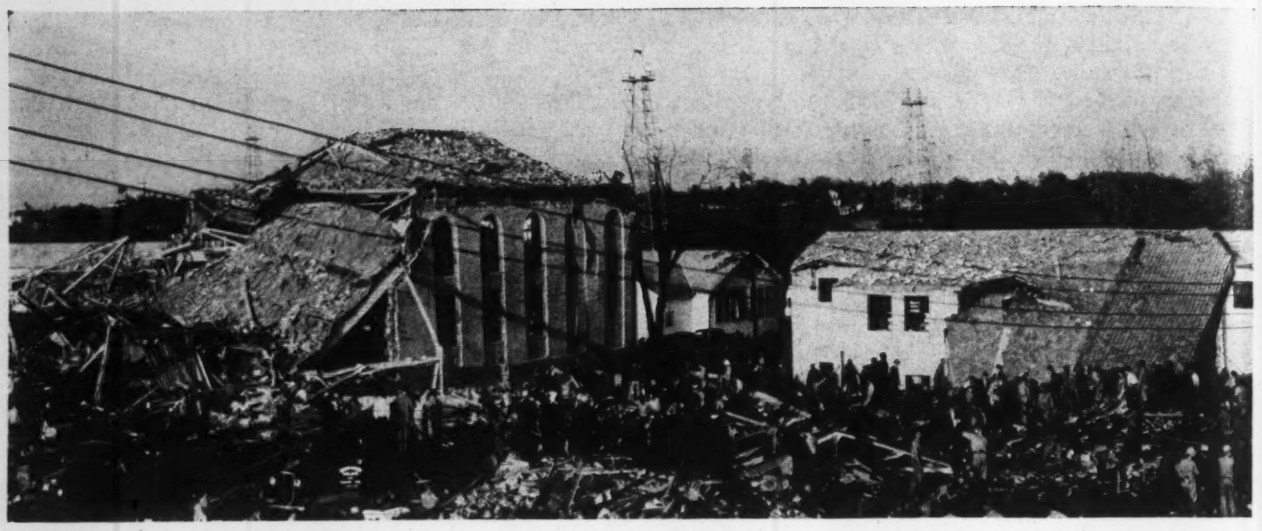


Fig. 8—Chart for finding the percentage of liquid by weight in a mixture of "Freon" liquid and vapor. This chart, together with the *A.S.R.E.* tables of the properties of "Freon," facilitates the computation of the density of "Freon" mixtures with a higher degree of accuracy than can be read from Fig. 3. However, for most practical purposes, Fig. 3 is satisfactory



Investigations by Federal Bureaus of Texas School Explosion Summarized

A SUMMARIZATION of the conclusions of specialists of the United States Bureau of Mines detailed to investigate the cause of the disastrous explosion at the New London, Texas, Consolidated School Building on March 18, which resulted in the death of 455 persons, was given in a letter addressed to Hon. James V. Allred, Governor of Texas, by Acting Secretary of the Interior Charles West. The recommendations of the Bureau of Mines experts as to precautionary measures to be taken in the future were also incorporated in the letter, which follows:

The duties of the United States Bureau of Mines, Department of the Interior, include investigations of causes of gas and dust explosions and issuance of recommendations that will aid in preventing their recurrence. Accordingly, four experienced specialists of the Bureau's staff were assigned to a study of conditions relating to the deeply regretted disaster at the New London, Texas, Consolidated School Building on March 18, 1937. These men reached the scene of the explosion on March 19, started their studies on March 20, and continued them through March 27.

The Bureau's representatives were D. J. Parker, Salt Lake City, Utah, a mining engineer with years of experience in the investigation of gas and dust explosions not only in and around mines but also in surface structures; G. W. Jones, Pittsburgh, Pa., a recognized authority on the theory of gas explosibility who has had wide experience in the investigation of explosions in sewers and in other enclosed spaces; and H. B. Hill and Gustav Wade of the Bureau's Petroleum Field Office at Dallas, Texas, both of whom are thoroughly familiar with the behavior of petroleum and natural gas and the technique of their proper handling.

Although their final report has not been completed, these specialists have sent their conclusions and recommendations to the Director of the Bureau of Mines, and in turn, the Department is pleased to offer them to you. We hope that the following resumé of the Bureau's findings may be of value to the people of Texas and to the whole country in efforts to prevent catastrophes of the kind that took place at New London:

(1) The immediate cause of the disaster was an explosion of a gas-air mixture, resulting from an accumulation of natural gas in an unventilated space beneath the first floor of the main school building. This space extended along the west front of the building and had a volume of approximately 46,000 cu ft. The physical evidence shows that the source of ignition was near the north end of this chamber and that the explosion propagated in a southerly direction.

(2) The manipulation of an electric switch in the manual training room near a door leading into the unventilated region ignited the gas.

(3) The gas was present because of leakage from gas lines and their fittings suspended beneath the floor, with virtually no method of detecting the gas or of removing it by natural or forced ventilation.

(4) The gas did not seep through the soil; this fact was determined fully by evidence secured from 71 test holes that were drilled to the rock below the surface formation. Numerous tests were made of the atmosphere in the holes by means of detectors for inflammable gas.

(5) The great violence of the explosion probably was due to the existence of a large volume (possibly as much as 46,000 cu ft) of an explosive mixture of gas and air together with the fact that the ignited gas was in an enclosed and substantially confined space with little or no opportunity for expansion at low pressure. Hence maximum opportunity was afforded to build up a high pressure as the explosion wave progressed, resulting in wide-spread disrupting and destructive effect.

(6) The results would have been the same under identical conditions regardless of the composition of natural gas used.

Recommendations of Specialists

The following recommendations have been made by specialists of the Bureau of Mines who have studied the evidence of the New London disaster:

(1) Public buildings should not have unventilated, relatively inaccessible, enclosed spaces, especially in the basement region.

(2) Basements, attics, and all spaces below floors, steps, and similar places in public buildings or buildings in which the

public assembles should be provided with adequate, constantly functioning ventilation.

(3) All gas and electrical lines serving public buildings and other buildings where the public assembles, and all piping, connections, fixtures, appurtenances, and appliances in such buildings, should be of permanent construction, placed by competent persons in accordance with the latest approved fire underwriters' or other applicable specifications, and operated in accordance with known safe practices; also, they should be kept in repair at all times.

(4) Responsibility for the construction, maintenance, and operation of gas pipe lines serving the public, and especially for the regulation of pressures in such lines, should be exclusively under the control of persons or organizations experienced in such work. Responsibility for the testing, lighting, and general care of the entire heating system should be entrusted only to persons who are familiar with such systems and competent to handle them safely and efficiently.

(5) Advantages in safety may be gained by placing gas mains or headers outside of buildings (especially public buildings and buildings where the public assembles); thereby minimizing the hazard of gas leakage in enclosed spaces. In some instances weather conditions, architectural design, or possibilities of aggravated corrosion may make such construction inadvisable. However, a desirable objective is to reduce to a minimum the enclosed piping for fuel purposes within a public building. The explosion in the New London School probably would not have occurred if the individual connections to the header had been made outside the building and the service lines brought into the building through risers from the header.

(6) Although the Bureau's tests proved that the explosion at the New London school was attributable in no way to seepage of gas through the soil, experience of the Bureau of Mines with other explosions indicates that gas may enter basements and other enclosed spaces that are partly underground. This possible hazard may be present especially where subsurface conditions permit gas to travel laterally through the soil along pipes that enter buildings below the ground surface. Accordingly provision should be made to seal all openings where pipes pass through walls.

(7) Central heating systems, with the heating plant isolated in a separate structure, have definite advantages with respect to protection against fire, explosions, and carbon monoxide. They also have certain advantages in efficiency with respect to large public buildings.

(8) Suitable warning agents with distinct odors, introduced into fuel gases that have little or no odor of their own, have proved to be of great aid in detecting leaks in public buildings and private homes. Even though the teachers and pupils of the New London school were accustomed to odors of gas and oil from the oil field in which they lived, it is likely that a malodorant in the gas used at the school would have given enough advance warning to prevent the explosion. Although use of malodorants is only one factor to be considered (as explosions have occurred with mixtures of air and manufactured gas which has a distinct odor) study should be made of fuel-service installations with the view of determining the need for impregnating the gas with a warning agent that will enable users of the gas to detect leakage. Warning devices based on instruments that will detect presence of inflammable gas and give an alarm also are to be considered, but obviously if such systems are to be effective they must be maintained carefully and tested frequently.

(9) There is need for a State supervisory body to inspect structures and buildings in which the public assembles lying outside the corporate limits of cities and towns where municipal safety inspection is in effect.

(10) Careful inspection by State or municipal agencies should extend not only to the construction of public buildings or buildings in which the public assembles but also to plans and design of such buildings. This safety inspection should include structural features as well as the details of heating and ventilation.

The use of warning agents for odorizing gas was developed in research by the Bureau of Mines. If you should desire information on this subject, the Department will be pleased to detail technical personnel experienced in this field to cooperate with you. The Department also offers other facilities that may be available for increasing safety to life and property within the State of Texas.

Dr. Price's Report

A report on the explosion by Dr. David J. Price, principal engineer in charge, Chemical Engineering Research Division, Bureau of Chemistry and Soils, U. S. Department of Agriculture, has been printed by the Government Printing Office, Washington, D. C., as Senate Document No. 56. The report describes the building, the method of heating by gas-steam radiators, the gas used for fuel, and presents Dr. Price's conclusions as to the cause. These were summarized in a press release issued by the Department on March 29 as follows:

In his report Dr. Price states that in his opinion the explosion was due to the ignition of an accumulation of combustible gases in an open area underneath the first floor of the building by a flash from electrical equipment in connection with the operation of a sanding machine located in the manual training shop. Price recommended the following precautionary measures for prevention of explosions in public buildings and institutions:

(1) Requirement for use of effective malodorants for detection of escaping combustible gas due to leaking equipment or other causes.

(2) Determination of practical application of gas indicators to detect presence of escaping combustible gases at percentages below the lower limit of explosibility and development of alarm systems and warning devices for schoolhouses and public buildings.

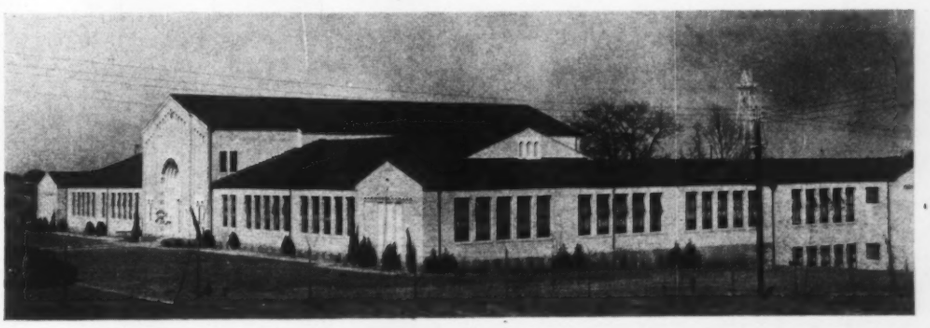
(3) Provision for supervision and inspection of public buildings and installations of heating and lighting devices and equipment.

(4) Installation of all electrical equipment and appliances in accordance with requirements of National Electrical Code.

(5) Provision for proper construction of buildings to prevent dead storage space underneath class recitation rooms and similar meeting rooms where dangerous gases can accumulate.

(6) Provision for adequate ventilation of all necessary spaces under all occupied sections of public buildings.

(7) Provision for adequate ventilation at all times for schoolrooms using gas heating appliances.



(The photographs reproduced here were made available through the courtesy of the International Union of Operating Engineers, Local 143.—Ed.)

Piping for 3500 Lb Steam

By H. L. Solberg*

The piping for the experimental steam generating unit at Purdue University, which can be operated at any desired pressure up to 3500 lb per sq in. and with a maximum steam temperature of 830 F, is described, and the experience with the piping joints is briefly reviewed. . . . The set-up is being used for studies of the design of forced-flow steam generating units for high pressure service

IN 1931, an experimental once-through, forced-flow steam generating unit was installed in the mechanical engineering laboratory at Purdue University. This unit, which has been described elsewhere (see *A.S.M.E. Transactions* for November 15, 1932), can be operated at any desired steam pressure up to 3500 lb per sq in. and with steam temperatures up to 830 F. The maximum steam temperature is limited to 830 F by the low carbon steel superheater tubes. The use of alloy steel superheater tubes would permit higher steam temperatures. The steam generator contains approximately 1800 linear feet of 1 in. O.D. x ½ in. I.D. low carbon seamless steel tubing welded by the oxy-acetylene process into two parallel circuits. There are no drums or heavy forged parts and the all-welded construction results in a one piece boiler which is continuous from feedwater inlet to superheater outlet without hand-hole plates, clean out plugs or any kind of joint which can be opened.

The steam generator is fired by a single mechanical atomizing oil burner and is provided with complete automatic control based on the use of selsyn motors and thyratron tubes. Steam output is regulated by hand manipulation of a throttle valve which, in a commercial installation, would be the speed governed valve on the prime mover. Normally, about 40 minutes are taken to bring the unit up to operating conditions and place it on automatic control, although it can be done in 15 or 20 minutes and perhaps even less by using an oversize oil burner tip. In shutting down, the fuel valve is closed and the feed pump is maintained in operation to provide a flow of water through the circuits until the setting is cool. Within five minutes after closing the fuel supply valve, the boiler gage will register the friction through the circuits due to the flow of water. In other words, the temperature in the steam piping is reduced about 600 F in five minutes on shutting down.

While it is comparatively easy to design a piping system for high pressures and *low* temperatures, the combination of high pressures and *high* temperatures presents a serious problem which is rendered more complex when the temperatures are subject to rapid changes. The few difficulties which have been encountered in the high pressure piping system can be attributed principally to rapidly fluctuating temperatures.

While the circuits of the steam generator were made of low carbon steel, it was felt that a better grade of material should be used in the piping system because of its proximity to the heads of the operators. The high pressure steam piping was therefore fabricated from double extra heavy seamless steel tubing having the following analysis:

Carbon	0.15 per cent
Manganese	0.6-0.8 per cent
Chromium	0.85-1.05 per cent
Molybdenum	0.3 per cent

The maximum size of steam pipe in the installation is 2 in. and the remarks made in this discussion are therefore confined to experiences with relatively small pipe sizes.

Screwed Connections

The high pressure piping system contains about a dozen screwed tees and ells of 1½ and 2 in. size. These are forged steel 6000 lb standard hydraulic fittings. Two of them were screwed up tight, subjected to a hydrostatic test and then seal welded by the electric arc process. Upon subjecting them to a second hydrostatic test, it was discovered that the welds were not sound and the joints leaked badly. The welds were made without preheating the pipe and fitting, and it is believed that the unequal expansion of the heavy fitting and thick walled pipe during the welding process caused the pipe to take a permanent set upon cooling, resulting in a loose set of threads. Perhaps this difficulty could have been avoided by a welder who had had considerable experience with this type of work. Since the only object in back welding was to seal the joint and the result was

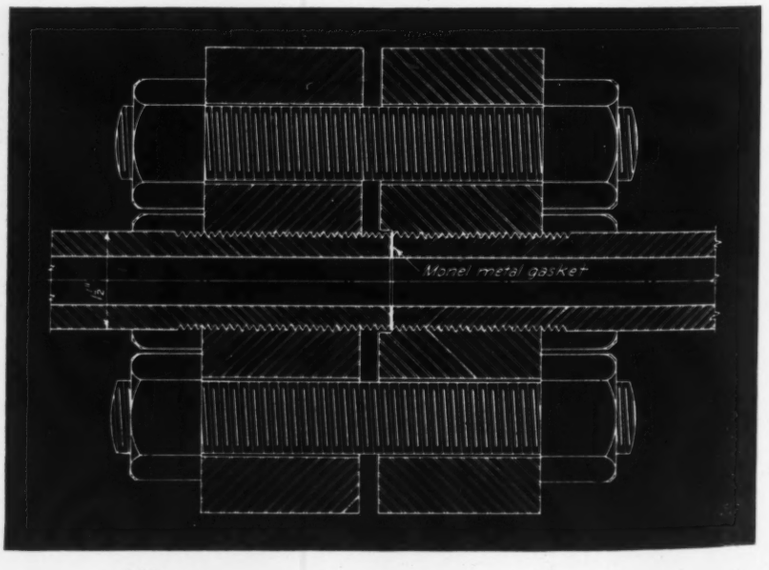


Fig. 1—Flanged joint for temporary experimental installation

*Professor of Mechanical Engineering, Purdue University.

a loose set of threads, the other screwed joints were made up tight with a commercial thread lubricant without seal welding. They have never given any trouble with leakage.

When some of the screwed connections were broken at a later date in order to change an experimental set-up, it was found that the threads were frozen and the joints could only be opened by stripping the threads from the alloy steel pipe. Incidentally, the threads in the fittings were not damaged. Subsequent experience with screwed connections on pipe containing steam at temperatures up to 1100 F has shown that many thread lubricants are unsuited for high temperature service where the joint must be opened occasionally. Dry graphite or graphite mixed with a small amount of red lead and water has been used with some success, but thread lubricants containing hydrocarbon oils have caused trouble. There is at least one commercial thread lubricant on the market at present which has been very satisfactory when subjected to these high temperatures.

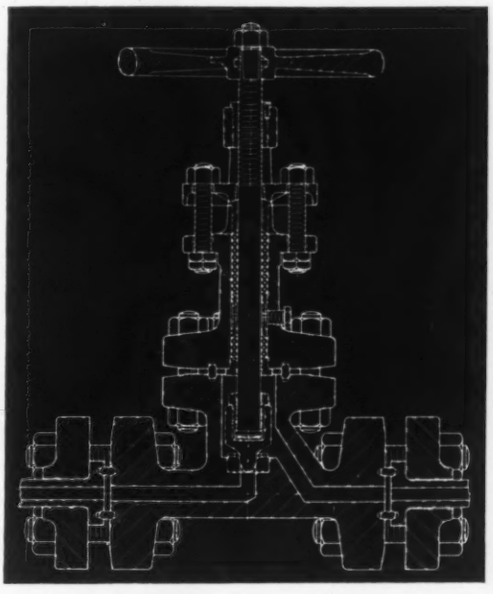


Fig. 2—Three-fourths inch throttling valve with ring joints

Flanged Connections

In normal power plant service at high pressures and temperatures, the creep strength of flange and bolting material is of fundamental importance. In this experimental installation, which is operated intermittently, the effects of rapid temperature changes during starting and stopping periods are of more importance than creep during steady temperature operation. When high temperature steam is admitted to a cold line, the pipe and inner parts of the flanges are heated more rapidly than the bolts, thereby increasing the compression on the gasket and the stresses in the bolts because of unequal expansion. If the bolts have been improperly tightened when cold, they may be stressed beyond the elastic limit, or the flanges or gasket may be deformed during the heating process. Conversely, during a rapid reduction in steam temperature, the pipe and inner parts of the flanges are cooled more rapidly than the bolts with the result that the bolting pressure on the gasket is reduced and the joint may leak.

Leakage due to unequal expansion caused by rapid temperature changes was encountered in the flanged joint illustrated in Fig. 1. This type of joint was used on a temporary installation where it was desired to measure the head loss through several tubes of different internal diameters. To facilitate interchange of tubes, all tubes had the same external diameter. Interchangeable parallel threads were cut on the tubes to permit screwing on of flanges with loose threads. The ends of the tubes were faced in a lathe and butted together with a monel metal gasket. The flanges were sawed from a piece of scrap car axle steel and were made 2⅜ in. thick so that they would not be deformed by the bolt load. Four 1⅛ in. alloy steel bolt studs were used in each joint. The joint was always tight during starting and normal operation but invariably opened and allowed steam to blow during the shutting down period. While annoying, this trouble was not sufficiently serious to prevent the use of the joints. No amount of tightening of the bolts when cold would prevent the leakage. It is probable that during the heating operation, the temperature of the pipe increased much faster than that of the bolts and the gasket pressure became so high that the material deformed, while on stopping, the cooling of the pipe reduced the gasket pressure to a point where the joint opened. This difficulty could only be overcome by manipulating the fuel feed and throttle valves on shutting down so as to maintain the steam temperature above 700 F until the pressure had dropped to less than 500 lb per sq in.

Another flanged joint, of the tongue and grooved type with a monel metal gasket, also gave some trouble with occasional leakage during the shutting down period. Inasmuch as the gasket seating surface in this type of joint is in the flanges instead of the pipe itself, this joint was less sensitive to rapid temperature changes than the type shown in Fig. 1. From the standpoint of minimum unequal expansion due to temperature changes, the gasket seat should be in the flanges as close to the bolt circle as possible.

Fig. 2 shows a ¾ in. throttle valve constructed with ring joints. The high pressure lines contain a number of ¾, 1½ and 2 in. valves and companion flanges which use the ring joint illustrated in Fig. 2. The gasket is a solid ring of soft steel, elliptical in cross section, which is retained in flat bottom V grooves machined in the valve flange and upset end of the pipe. Rapid fluctuations in pressure and temperature have not caused these flanges to leak. This is attributed to the fact that the soft iron gasket is seated in the flanges with a wedging action at a sharp angle. Consequently, a given differential expansion between the bolts and the flanges at the point where the gasket seats will have much less influence on the gasket pressure than when the conventional type of gasket is used with its flat seating surface perpendicular to the center line of the bolts.

In the opinion of the writer, high pressure, high temperature steam piping systems should be designed in such manner as to reduce the number of flanged connections to a minimum and to substitute welding wherever possible. Flanged joints are available which are satisfactory even on rapidly fluctuating temperatures, at least in the smaller sizes. However, there is room for designers to exercise ingenuity in developing some form of joint which can be opened without too much effort but which does not require the amount of metal and labor which now go into flanged joints of conventional design.

E. Vernon Hill Comments . . . .

THE OPERATING ENGINEER has much to do with the satisfaction afforded the owner by an air conditioning installation.

Last summer we talked with the owner of a large recreation building cooled by a refrigerating plant; he complained of unsatisfactory conditions and expressed the opinion that the cooling capacity was entirely inadequate. We then talked with the engineer; he advised us that he was getting the last ounce of capacity the apparatus would deliver. "I keep the suction lines well frosted," he said, "and the entering condenser water at 60 F."

He accomplished this by using a generous mixture of chilled water with the city water supplied to the condensers—of course robbing the air cooling coils in proportion.

The engineer was convinced, with some difficulty, that he was *not* getting the last ounce of capacity out of the plant and with corrected operating conditions this was easily demonstrated, to the satisfaction of the owner.

A similar experience, earlier in the year, may be of interest. It was a theater building with a spray cooler and an 85 ton CO_2 compressor driven by a 100 hp motor. A hot spell in May required the operation of the cooling plant. With the complete system in operation we observed the temperature of the spray water to be 49 F and the temperature of the city water to the condenser, 47 F. We advised the engineer that if he would feed the city water at 47 F directly to the washer he could do more effective cooling and incidentally save wear and tear on his compressor, and the cost of operating a 100 hp motor. However, we left him skeptical and unconvinced.

While it is unsafe and unwise to draw conclusions from occasional experiences of this kind, nevertheless we have a feeling that a good many cooling plants that have been in service for some time could be made to deliver greater capacity and give better satisfaction.

It is easy to blame the operating engineer but operating engineers as a class are measuring up to the new requirements better than would be expected, considering the constant changes and improvements in practice that occur almost from day to day.

We are inclined to think that contractors who are installing equipment are partly to blame. It is not enough to tell the operating man what switch to throw, what valves to open and close when he stops and starts the equipment. He should have very complete information, particularly the fundamentals without which he can never meet unusual conditions. He should be taught to sleep for a while (we are speaking figuratively of course) with the thermodynamics of the refrigeration cycle and the psychrometric chart.

The owner also should be advised regarding the importance of correct operating methods of procedure and this is particularly true when for any reason the operating engineer is replaced by a new man.

The successful and economical operation of a cooling plant is not guaranteed indefinitely by the good reputation of the manufacturer or the contractor, so much depends on the knowledge and ability of the operating engineer.

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SERVICE ENGINEERS, the kind that know their stuff, if we may be permitted this slang expression, are coming more and more into the picture. The larger manufacturing and contracting concerns usually employ the best service engineers, in fact, they seem to have a corner on the market, and a free lance service engineer with ability and experience is hard to find. We are inclined to think here is an unexplored field that might be cultivated with profit.

The service engineers in the employ of manufacturers or their contracting representatives are necessarily handicapped by their connections and can as a general proposition only service their own installations. A good service engineer, on his own so to speak, would find a fertile field in correcting bad conditions in existing plants and modernizing some of the old equipment that has been in service for a good many years.

It sounds rather strange to speak of modernizing air conditioning equipment but one who investigates will find modernizing in order in many plants only a few years old.

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CONSULTING ENGINEERS, and we refer specifically to the older type who knew all and did all after a fashion, are apparently in eclipse. They were accustomed to assume the responsibility for the heating and ventilating of a building, the lighting system, sprinklers, electricity, and sometimes also the plumbing. The broadening equipment field, however, has resulted in these numerous and diverse activities being more or less split up and spread around. It is the age of specialists and the engineer who can handle a complete air conditioning installation competently, in all its various aspects, has his hands full.

In fact, in this field the individual engineer operating in a consulting capacity usually requires the assistance of others specializing in its different branches.

The multiplicity of activities has given rise to a condition that is rather confusing and new designating terms for engineers are appearing in the literature. We hear of designing engineers, application engineers, service engineers, as well as refrigerating engineers, heating and ventilating engineers, operating engineers, and many others, some of them new terms that never have been well defined, and the meaning of the old terms is rapidly changing.

It is somewhat confusing and we suggest that the *A.S.H.V.E.* could render a real service to the profession and the industry by adopting a definite authoritative classification of engineers and their respective fields of operation.

Formulas for Stresses in Bolted Flanged Connections

By E. O. Waters*, D. B. Wesstrom**, D. B. Rossheim†, and F. S. G. Williams††

Progress in pressure vessel design and fabrication during the past two decades has brought about a marked change in the form of bolted connections. Whereas flanges were formerly provided with hubs of approximately uniform thickness, screwed to or slipped over the shell or pipe, it is now customary to use tapered hubs, and an integral structure is obtained by butt welding the end of the hub to the shell. This has invalidated former methods of design for flanged connections. . . . . This paper outlines a revised analysis based on the ring, tapered hub, and shell being considered as three elastically coupled units loaded by a bolting moment, a hydrostatic pressure, or a combination of the two. Design formulas and charts are given for the computation of stresses that are likely to be critical, and their application is illustrated by three problems which are taken from current commercial practice.

FOR the last two decades there has been a steady increase in the pressures and temperatures employed in the generation of steam and in the process industries. This development has been coincident with a rapid increase in the size of the units. Approximate methods of design that had previously been acceptable could not be used for high temperature, high pressure designs involving large equipment. With high temperatures and pressures, encroachment on safe operation could not be tolerated; and at the same time, the size of most units demanded economy in design without any compromise with safety. The result has been a steadily increasing need for flange stress formulas which are correct over a wide range.

During this period, methods of construction have also undergone radical changes. Earlier flange designs were nearly all based on flanges having hubs of approximately uniform thickness. The advent of welding has changed this and the most commonly used construction involves a flange with a tapered hub which is butt welded to a shell or pipe.

A number of different methods have been used for calculating the stress in a flange, which are reviewed later in this paper. The general observation can be made, however, that in each case they had limitations which confined their use to certain types or ranges of flange cross section.

Extensive interest in the subject dates back to about 1927 when the Taylor-Waters formulas were presented before the *A.S.M.E.* (1).[1] These formulas had been based upon, and checked by, actual deflection tests conducted by J. Hall Taylor in 1924 and 1925. This method of calculating the stress in a flange has been the one most widely used since that time for the reason that it was substantiated by test data and was reasonably accurate over a wide range.

Flange design became a subject of extensive discussion and the methods of approaching and handling the problem of practical design were numerous. This resulted in the appointment of a joint committee composed of representatives of the *A.S.M.E.* Boiler Code Committee, the Joint *A.P.I.-A.S.M.E.* Pressure Vessel Committee and *American Standards Association* Committee B16E. In 1934 this committee drafted a set of rules (2) which set up a uniform procedure covering all important phases of the design, and represented the first known effort to introduce a rational stress analysis into the commercial design of flanged connections. This committee is continuing its work in an effort to improve these rules and recently it has had the additional assistance of the *Heat Exchanger Institute*.

One of the most important actions of this committee was a recognition of the importance of the longitudinal bending stress in the hub. Prior to the publication of these rules most designs had simply checked the tangential stress at the inner diameter of the flange and had ignored the possibility of the hub stress being the critical factor. The formula for the hub stress appeared in the Taylor-Waters paper (1) of 1927 but was not generally used.

Because of the conditions mentioned, the authors wish to present in this paper a condensed discussion of the problem of flange stresses and their calculation. The various methods that have been used are summarized with a brief analysis of each and finally formulas are presented that are believed to give accurate results over a wide range.

Review of Earlier Work

Brief comments on some of the more important developments in the calculation of flange stresses will serve as a background for this paper.

Perhaps the earliest method of calculation to receive wide attention was the so-called *"Locomotive"* method, (3) generally credited to the late Dr. A. D. Risteen. The section *abcdefg* in Fig. 1 is assumed to rotate counterclockwise, but without distortion. The final equation is in effect the conventional flexure formula, the external moment being the total bolt moment per radian angle and the section modulus being that taken about the axis X-X' through the center of gravity O.

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[1]Numbers in parentheses refer to bibliography at end of article.

For ring flanges this gives the tangential stress on either face, and for hubbed flanges it gives the tangential stress at the free end of the hub.

Crocker and Sanford developed a method (1, 4) whereby the flange is analyzed as a beam, in which bending about the neutral axis X-X'' takes place on the section A-A, and the external loads are one-half the bolt

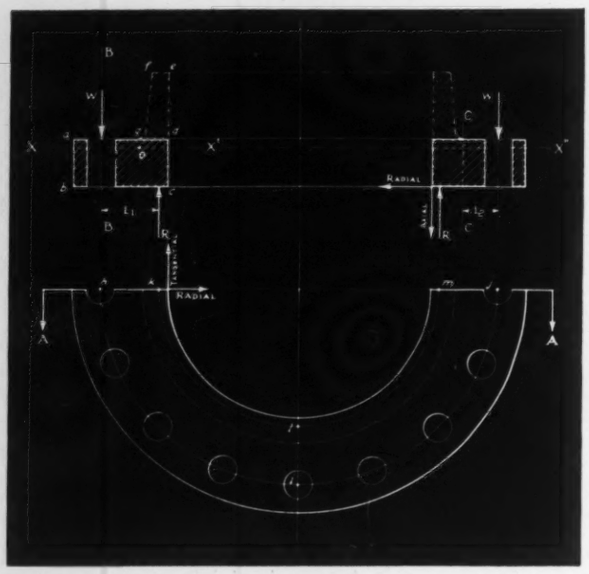

Fig. 1—Illustrations of earlier methods of calculating stress in a bolted flange

load W and one-half the reaction R, each concentrated at the center of gravity of their respective half-circles *hij* and *klm*. (The location of the bolt-load circle, however, was assumed tangent to the inner edge of the bolt holes, and not along the bolt circle as shown in Fig. 1.) This method likewise gives the tangential stress on either face of a ring, or at the free end of the hub.

Den Hartog (4) showed by vector analysis that although the *Locomotive* and Crocker-Sanford methods are derived in different ways, they are fundamentally identical.

A method devised by Tanner for ring flanges, and discussed by Waters and Taylor (1), is to assume the ring to be fixed at the section B-B around the bolt circle and to be equivalent to a cantilever beam of length L_1 with the "concentrated" load R uniformly distributed across a width equal to the circumference of the ring. This method gives the radial stress assumed to be present at section B-B. In the application of the method, Tanner took account of the tangential stresses by using suitable factors derived from experiments on rings of the proportions in which he was interested. The Tanner method was modified by Crocker (1, 4) for application to hubbed flanges (and presumably adaptable to ring flanges also) by assuming the fixed section to be the weakest section C-C in the ring at the base of the hub, with the load W "concentrated" at the distance L_2 at the free end and distributed along the bolt-loading circle. This likewise results in a calculation of the radial stress assumed to exist, in this case at section C-C.

None of the foregoing methods took into account all the conditions present in the flange under load, and so the Waters and Taylor paper in 1927 (1) based on a combination of the flat plate and the elastically supported beam theories, was probably the first instance in which the stress conditions in a flange in the three principal directions—tangential, radial, and axial—were explored with the object of determining the location and magnitude of the maximum stress. Formulas were included for the deflection of the ring, and the calculated deflections were compared with those actually obtained in several series of tests, the data of which were also reported. Because in flange proportions considered at that time the tangential stress in the ring at the inside diameter was the controlling factor, the formulas for stresses elsewhere in the flange were generally overlooked by designers.

The Waters-Taylor paper evoked extensive discussion (4) in the course of which Timoshenko presented an analysis for both ring flanges and hubbed flanges, including a method of dealing with hubs shorter than the so-called "critical" length. Most of these formulas can be found also in his work on "Strength of Materials" (5).

In 1931 Holmberg and Axelson wrote a paper (6) in which they used the flat plate theory in developing formulas for stresses in loose ring flanges and in flanges made integral with the wall of a pressure vessel or pipe.

In a series of articles recently published (7), Jasper, Gregersen, and Zoellner discussed further the formulas of Timoshenko, and Holmberg and Axelson. They also made an outstanding contribution to the subject by presenting the results of an extensive series of tests on plaster of paris models. Some of the data obtained were used in developing an analysis of the stresses in hubbed flanges having a large circular fillet at the junction of hub and ring.

When the rules for flanges in the *A.S.M.E.* and the *A.P.I.-A.S.M.E.* Unfired Pressure Vessel Codes (2) were first published in 1934, the wide range of their application made it necessary to use formulas based on a rational and complete theory, and because the Waters-Taylor equations met this requirement and had been checked by experiment, they were adopted, with auxil-

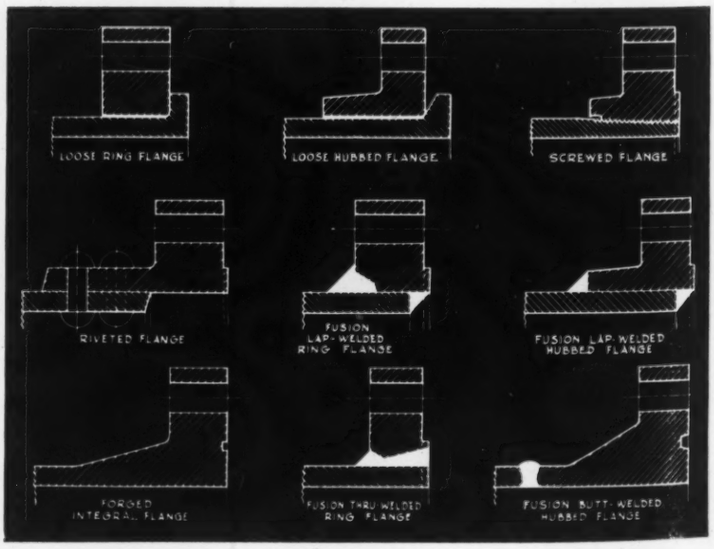

Fig. 2—Some types of flanges to which the authors' formulas are applicable

iary charts to simplify the calculations. The radial stress formula was omitted, however, because it was not believed that it would be the critical factor in any practical design.

Several years of experience with the *A.S.M.E.* rules and the rapid development of large size equipment for high pressures and temperatures have indicated the need for a revamping of these formulas. The influence of shearing forces at the junction of hub and ring was not included in their derivation, and this effect is important in many instances. The formulas are inconvenient to handle because they are based on the mean diameter and thickness of the hub; also, other assumptions on which they are based lead to difficulties. The method of handling hubs of non-uniform thickness, such as tapered hubs and large circular fillets, has proved unsatisfactory and it is believed does not give proper credit for reinforcement provided by such designs, where a small amount of metal is so distributed as to result in greatly increased strength.

In the sections that follow the authors develop a method of flange stress calculation which it is believed will be easier to handle than the previous form of the formulas, and which is accurate over the complete range of practical design covering hubs of either uniform or tapered section.

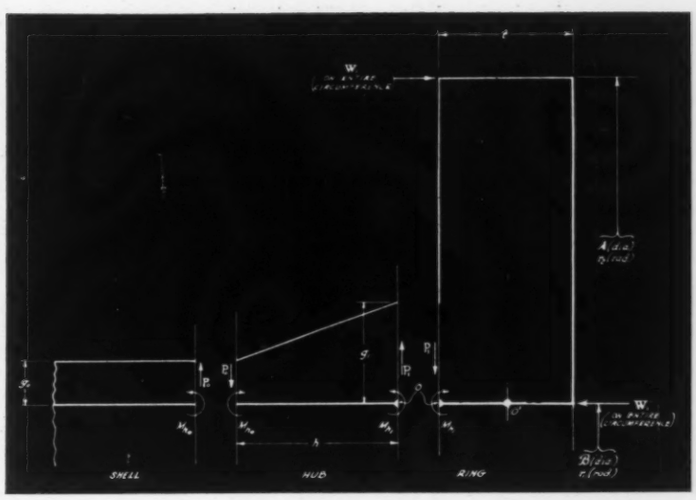

Fig. 3—Analysis of forces and moments in a tapered hub flange

Basis of the Present Study

The formulas presented in this paper can be applied to the circular flanges of bolted joints under pressure, such as flanges on heat exchangers, condensers, vulcanizers, and piping, when the flanges are free to deflect under the action of the bolt pull. This includes all types of flange facing where the gasket or contacting flange surfaces are entirely within the bolt circle, and excludes all cases where there is any contact outside of the bolt circle, as in a full face gasket.

It is assumed that the flange is made of a homogeneous material having stable elastic properties and that these properties obtain under the conditions of loading assumed in the design. It is beyond the scope of this paper to discuss what occurs when the material becomes plastic. However, the general statement may be offered that when using these formulas to design steel flanges for high temperature service, creep or progressive plastic deformation need not be taken into account if the design is such that the calculated stresses do not exceed the values at which creep will have an appreciable effect. In other words, the design should be based on the conditions obtaining in service after relaxation of the bolts and flanges is practically complete, using an appropriate safety factor.

It is not the purpose in this paper to deal with the probable stresses set up in the bolts on tightening them, or with the gasket or contact pressure required either to seal the contact surfaces or to maintain pressure tightness. These are separate items in the general subject of flange design and can be considered independently of the study of stresses in the flange under any given bolt load. It is assumed therefore that the bolt load has been determined, and this is used as a starting point.

It is also assumed that the lever arm of the bolt load has been determined. Most flange designs involve a compound lever arm, and a method, not theoretically exact but sufficiently approximate for practical purposes, for calculating compound lever arms may be found in Par. UA-20 of the *A.S.M.E.* Unfired Pressure Vessel Code or Par. W-317 of the Joint *A.P.I.-A.S.M.E.* Unfired Pressure Vessel Code (2).

The further assumption is made that for practical purposes the effect of the external moment on the flange, equal to the product of the bolt load and the lever arm, is the same in all cases, regardless of the location of the bolt circle and of the forces balancing the bolt load. This assumption may appear rash, but studies have been made that indicate it is practically true for ring flanges without hubs, and it is reasonable to suppose that the same would obtain for hubbed flanges. The equations in the paper by Holmberg and Axelson (6) enable a study of the effect of applying the external loads at various diameters of a ring flange, and a detailed analysis of these equations has shown that the points of application can be varied over wide limits without affecting the results to any practical extent.

It should be noted here that the formulas presented herein are applicable to flanges without limit as to diameter, pressure, and temperature, and aside from the restriction as to type of flange facing noted previously, to flanges of all types. Some of these types are illustrated in Fig. 2, in which obviously the various forms of flange facing may be interchanged. Thus the formulas may be applied to loose flanges, with or without hubs, screwed flanges and flanges attached by riveted or welded lap joints, and integral flanges and flanges made integral by through welding a ring or butt welding a hub. In the Unfired Pressure Vessel Code (2) screwed flanges are considered as loose flanges because the shell or pipe offers very little, if any, resistance to the deflection of the flange. Flanges riveted and welded by means of lap joints are also considered as loose flanges. This results in a heavier flange, which is desirable in order to keep the unknown tension stresses on the rivets and fillet welds at a reasonable minimum.

The following analysis considers the general case of a tapered-hub flange, in the course of which a loose ring flange, a loose flange with hub, and either a loose or integral flange with hub of uniform thickness are shown to be special cases of the general one.

Analysis of Stress Distribution

The tapered hub flange may be subdivided into three portions, which will hereafter be termed the ring, hub, and shell, as shown in Fig. 3. Each portion may then be studied as an independent unit, with undetermined boundary conditions (displacement, rotation, moment, shear) at the junction surfaces. When the portions are assembled, each of these boundary conditions on one side of any junction surface is identically equal to a corresponding condition on the opposite side; in addition, certain conditions are known at the free boundaries of the assembled flange. Hence, sufficient relations exist for the evaluation of all boundary conditions in terms of the external loading, the dimensions of the flange, and the elastic constants of the material. From the mathematical standpoint, the problem is then completely solved; practically, it remains to investigate certain stress distributions and, if possible, establish simplified formulas, so that the labor of checking critical conditions in existing designs, or determining the proportions of new designs, may be lightened as much as possible.

The analysis may be greatly simplified by making the following assumptions:

1. Due to axial symmetry, the complete flange may be replaced by a unit sector, *i.e.*, a sector whose plane faces have the form of Fig. 3 assembled, and include an angle of $1/r_1$.
2. The loading consists of (*a*) internal hydrostatic pressure acting radially on the bore of the flange and, through an assumed closure, in an axial direction as well; and (*b*) a moment acting on the ring, so distributed that it may be replaced by an equivalent couple applied at the inside and outside diameter of the ring.
3. Stretching of the middle surface of the ring, due to application of the ring moment, is negligible.
4. Dishing, or rotation, of the ring is small, so that the two load systems and their elastic effects are linearly related; and the complete problem may be solved by superposition.
5. When the ring moment is the only external load in action, point *O* shown in Fig. 3 has negligible displacement.
6. The hub and shell may be treated as membranes subject to tension and bending, and the radius to the middle surface may in general be used interchangeably with the inside radius r_1 shown in Fig. 3.

[To be concluded]

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Corrosion of Condensate Return Lines

J. H. Walker* Answers a Reader's Question, Suggests How to Go About Solving the Trouble

A READER of H.P.&A.C., director of buildings and grounds of a university, recently submitted the following corrosion problem, asking for suggestions as to the cause of the trouble, and a remedy.

"We are having considerable trouble with pitting and corrosion in our underground lines which bring condensate returns from the building heating systems back to the heating plant. Most of our trouble has been in the line from the gym. Recently we took out some extra heavy fittings which were entirely eaten through by the corrosion, and are trying to figure why we should have more trouble in the gym line than in any other line.

"There is a steam hot water heater in the building, and on two occasions the packing around the tubes has leaked which has allowed city water to leak into the steam space and, therefore, to be brought back to the boiler room in the return line. On one occasion this leak went on for several days before it was discovered.

"We think it possible that the mixing of the city water with the hot returns might be at the bottom of the trouble, as there seems to be no other reason why we should have more trouble with the line from this building than in the lines from others."

ANSWER—It is unwise to attempt to diagnose a case of corrosion such as this without first having accurate water analyses by means of which the corrosive agents can be identified. In general, oxygen and carbon dioxide are the most common sources of corrosion in steam heating return lines. The cold supply water entering the return lines could carry with it a considerable amount of oxygen, but if the leaks mentioned existed only for periods of several days as stated, it is unlikely that serious corrosion could occur during such a space of time. In a return system on one university campus with which the writer is familiar, corrosion was definitely traced to the presence of oxygen and when the return water was passed through deaerators located at strategic points the corrosiveness of the water, as indicated by the *N.D.H.A.* corrosion tester, was reduced to a negligible level.

The return lines from water heaters have shown more corrosion than those from building heating systems, due probably to the higher temperature, higher rates of flow, and possibly to other reasons not thoroughly understood.

A complete water analysis made by a competent chemist is the first step in an intelligent attack on such a corrosion problem. Methods of taking samples and making such analyses have been prescribed by the *National District Heating Association*. The effectiveness of curative methods which might be adopted can be readily ascertained by the use of the *N.D.H.A.* corrosion tester.

*Superintendent of Central Heating, The Detroit Edison Co.

Practical Standards for Air Conditioning Calculations

[Part 2]

By W. S. Bodinus*

Pan type outlets in National Broadcasting Co. studio

BASIC principles of psychrometry, an understanding of which is needed in calculating air conditioning capacities, were reviewed in Part 1 last month. In Part 2, we take up sensible heat, followed by a discussion of air distribution and temperature differential between the room air and the entering air.

Sensible Heat Principles

When a substance permits heat to be added to or subtracted from it without changing its physical state, its temperature rises or falls. The heat that causes this temperature change is called sensible heat, since it is indicated by a thermometer or can be felt by the hand. The formula $\text{Btu} = W \times (T_2 - T_1) \times C$ is standard for determining the quantity of sensible heat involved in the temperature change; (W = weight of substance; $T_2 - T_1$ = change in temperature; C = specific heat.)

Since air is the medium of heat pick up in air conditioning—or rather the substance used in absorbing heat by a rise or fall in temperature—we have a formula for comparing temperature rise with quantity of air calculated:

$$\text{Cfm} = \frac{\text{Btu} \times 56}{60 \times (T_2 - T_1)}$$

Cfm = cubic feet of air circulated per minute,
Btu = heat absorbed per hour,
60 = minutes per hour,
T_2 = entering air temperature, F,
T_1 = leaving air temperature, F,
56 = cubic feet of air raised or lowered 1 F by 1 Btu.

The quantity of air is expressed in cubic feet per minute rather than pounds, and we use the standard weight of air (0.075 lb per cu ft, equal to 13.33 cu ft per lb) and an average specific heat of air of 0.24. Therefore $13.33 \div 0.24 = 55.5$ cu ft of air is raised 1 F by 1 Btu. The value "55.5" is generally taken as 56 and the basic air formula derived as follows:

$$\text{Btu} = W \times (T_2 - T_1) \times C$$
$$C = 0.24$$
$$W = 0.075 \text{ lb per cu ft of air}$$
$$\text{Btu per min} = 0.075 \times \text{cu ft per min.} \times (T_2 - T_1) \times 0.24$$
$$\text{Btu per hr} = 0.075 \times \text{cu ft per min.} \times (T_2 - T_1) \times 0.24 \times 60$$

$$\frac{\text{Btu per hr}}{0.075 \times 0.24 \times (T_2 - T_1) \times 60} = \text{cu ft per min}$$

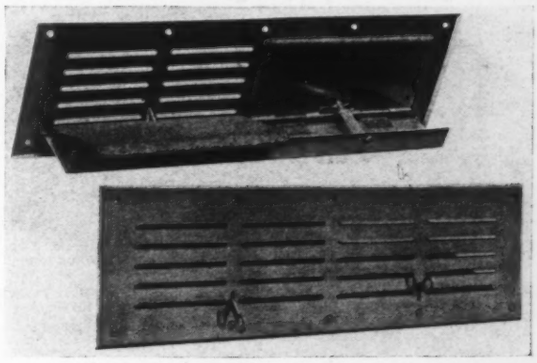
$$\frac{\text{Btu per hr}}{0.018 \times (T_2 - T_1) \times 60} = \text{cu ft per min}$$

$$\frac{1}{0.018} = 55.5 \text{ (say 56)}$$

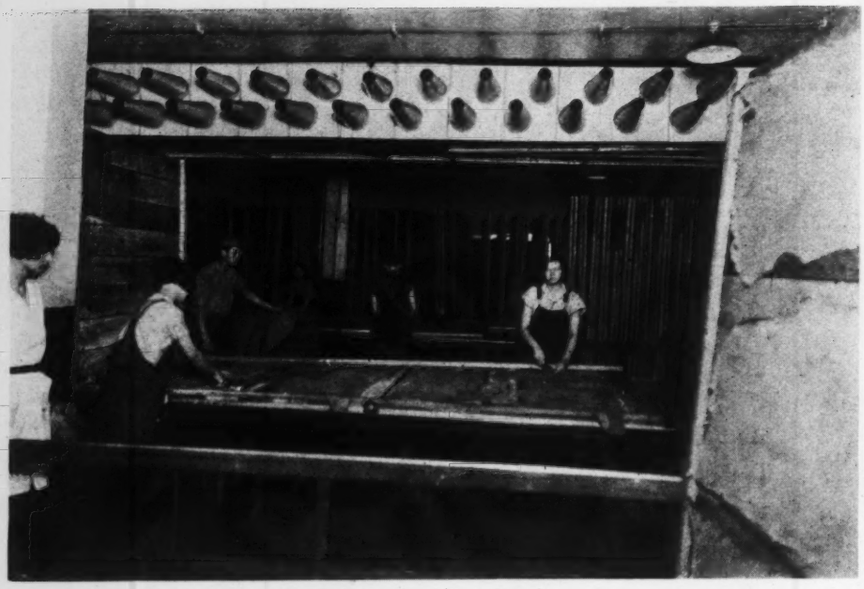
$$\frac{\text{Btu per hr} \times 56}{(T_2 - T_1) \times 60} = \text{cu ft per min}$$

Thus it can readily be seen that the quantity of air circulated per minute is determined from the sensible heat to be absorbed and the temperature rise or temperature drop of the air, depending on whether the

Slotted outlet with snap damper control, for high velocity discharge of conditioned air. Above: Rear view. Below: Front view



*Engineer, Carrier Corp.



Industrial application of ejector nozzle air distribution

problem is heating or cooling. The building heat losses are calculated on a sensible heat basis, which establishes the quantity of heat to be picked up. The allowable temperature rise is determined from analysis of the space to be conditioned, type of air distribution system and kind of outlets used. Temperature differentials in heating may be from 5 to 75 F and in cooling from 5 to 30 F, the final temperature difference in each case being established from experience as well as from data on the type of air distribution system used. If, for instance, the sensible heat load of a space is 150,000 Btu per hr and the temperature differential for summer cooling is 15 F, we would have to circulate

$$\text{Cfm} = \frac{\text{Btu per hr} \times 56}{60 \times (T_2 - T_1)} = \frac{150{,}000 \times 56}{60 \times 15} = 9333$$

Air Distribution and Temperature Differential

The most difficult problem in determining the quantity of air to be circulated is the temperature differential, especially for cooling calculations. Comparatively few performance tests on outlets and grilles have been made to ascertain the relationship between length of air blow, temperature differential, quantity of air supplied and minimum ceiling height.

Illustrated here are two types of grilles, one the slotted grille and the other the diffusion or adjustable grille. A typical condition taken from the manufacturer's data tables for a slotted outlet is as follows:

Grille No.	Velocity Fpm	Static Pressure In.	Cfm	Length of Blow Ft	Temperature Min. Ceiling Height	Temperature Differential F
35C1—5x5x⅛"	1500	0.14	117	12	8′ 0″	15

For the diffusion outlet a typical condition is:

Grille No.	Velocity Fpm	Static Pressure In.	Cfm	Length of Blow Ft	Temperature Min. Ceiling Height	Temperature Differential F
35C5-263	500	0.02	232	17	12′ 0″	20

With such information available on the outlets the only problem left in application is one of practical consideration; to avoid blowing against beams and columns and to space outlets close enough to obtain even distribution across the entire conditioned space.

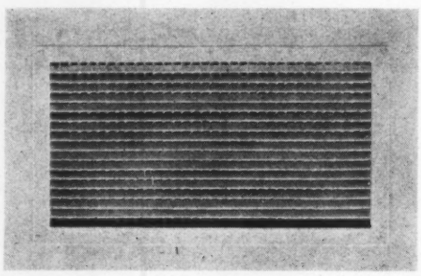
In establishing supply grille ratings of the slotted type of outlet illustrated, data were taken from test readings of many industrial installations such as the one pictured showing the nozzle or ejector system. The distribution of air for industrial product conditioning seriously affects the rate of drying as well as the quantity and uniformity of the product, so the data obtained from industrial applications give a good indication of the results necessary for comfort work. Similarly data for the diffusion type of grille have been obtained from industrial installations for product as well as comfort cooling. These grilles are used for greater lengths of blow with wider spacing between outlets, where large quantities of air are to be thoroughly diffused horizontally.

The pan type of outlet has been used extensively for many years and the tabulated data on these have been quite well established for some time. A typical example is:

Pan Size In.	Duct Opening In.	Velocity Fpm	Pan Drop In.	Cfm	Length of Blow Ft	Min. Ceiling Height	Temperature Differential F
20 x 20	10 x 10	300	4	335	10 (both ways)	10′-0″	12

Pans may be made of different materials, such as sheet metal, ply metal, plaster, wood, glass, etc.

No formulas are available at present to determine the exact size of grille for a given problem but rather manufacturer's tables must be used. These tables are becoming more and more accurate and complete as experience broadens.



Adjustable or diffusion type air outlet

After the sensible heat load has been calculated and the type of distribution system determined, the supply air quantity can be established. The amount of air in cubic feet per minute supplied to the room is then directly proportional to the quantity of heat absorbed in maintaining the room temperature. If the calculated maximum quantity of heat absorbed is less (or possibly, on a few occasions, more) the temperature differential must be increased or lessened to cause the air to pick up more or less heat. In most cases the quantity of air must remain constant to maintain an even distribution of air and proper length of blow to avoid cold drafts, and the control should generally be designed to modify the supply temperature. *[To be continued]*

X-Ray Inspection of Field Welded Pressure Piping

By **Herbert R. Isenburger***

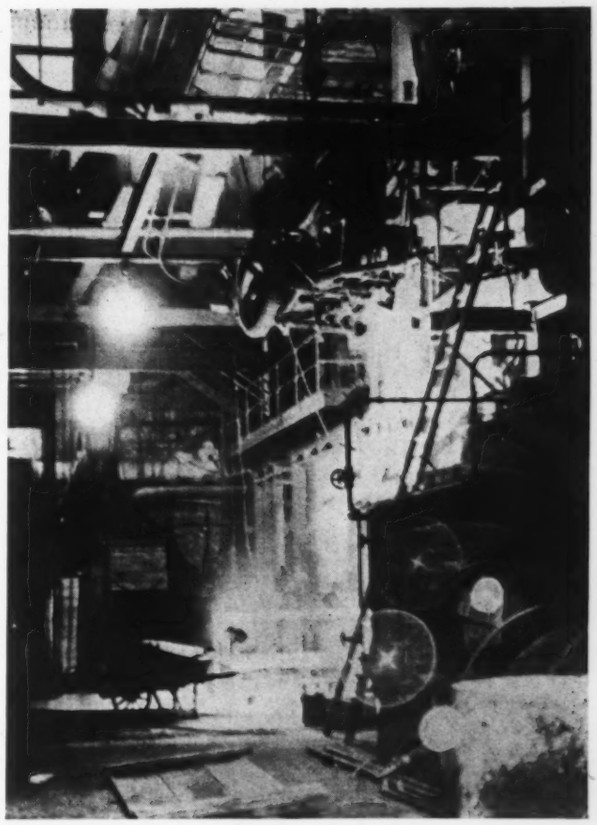
PIPE lines which connect boilers and turbines must be considered part of the complete system and their fabricating requirements should be similar to those prevailing for the boiler and turbine. Boilers are made under shop procedure which, as far as x-ray inspection is concerned, assures simple and effective examination of the welded seams. Most pipe joints have to be welded in the field, sometimes under difficult conditions. It certainly would seem to be highly important to x-ray these field welds.

Is such examination practically feasible? The answer can be found in the positive result of the first job of this nature. The writer and his assistant drove from New York to Omaha, Neb., carrying a portable x-ray unit on a trailer chassis. After a suitable field darkroom was installed on the job, we immediately started operations. It was by no means an easy matter to bring the x-ray machine to the various joints. In some cases the trailer was hanging on a chain block during the exposures and tube stand and tube were placed on scaffolding. More than 95 per cent of all x-rayed seams were covered on this job. The tube used was of special construction, air-insulated and water-cooled, as oil-insulated and oil-cooled tubes are not suitable for this kind of work.

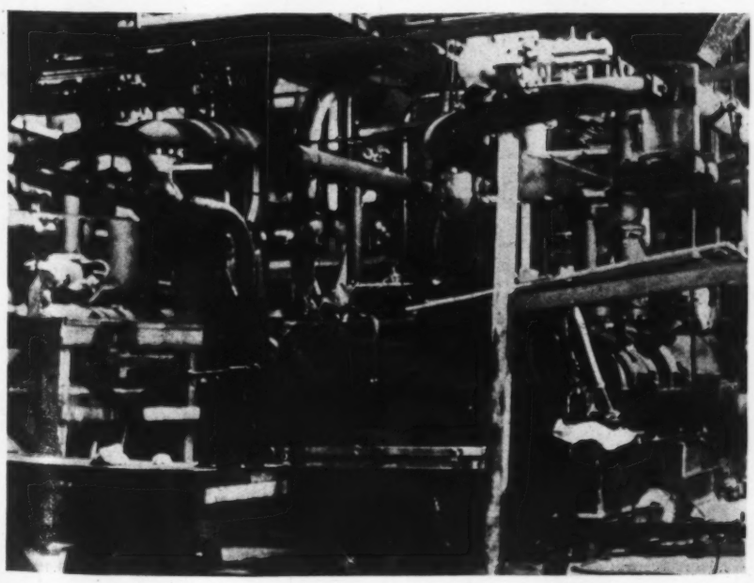
The x-ray technique employed involved the penetration of both walls of the pipe. The results obtained were most satisfactory. As to cost, it was reasonable and well worthwhile; the superintendent stated that the detection of one faulty joint would more than pay for the expense of x-raying the entire pipe lines.

It is interesting to note that the joints which were made under the most difficult circumstances showed up better than easier ones. As a matter of fact, the simplest joint required the most repair work. Only in one case was it found necessary to replace the entire weld.

Not long ago a prominent engineer stated that x-ray inspection of field welded piping was practically impossible. The project described demonstrates, I believe, that it can be done. With proper technique and interpretation, x-ray inspection blazes the trail for important welding work, and offers a means of proving the soundness of the weld.



Set-up for x-ray inspection 30 ft above the ground



A tight place for an x-ray machine to squeeze through

The modern covered wagon: portable x-ray equipment

*President, St. John X-Ray Service, Inc.

Low Pressure Exhaust Steam Chills Water for Building Air Conditioning

IT has been pointed out in these pages a number of times that there are many ways of air conditioning a building, and that the selection of the proper method for a specific case depends, therefore, upon a thorough engineering analysis of the requirements and existing conditions. An example is the installation at the Otis Building, Chicago, which is just starting its second season of service. Refrigeration for air conditioning is provided by a steam jet unit operating on 5 lb exhaust steam. There are numerous steam jet air conditioning systems using high pressure steam, but it is not widely appreciated that low pressure exhaust steam is adaptable to such service.

Building Generates Own Power

The Otis Building generates its own electric power, and is equipped with four boilers with a rated capacity of 41,400 lb per hr of steam at a pressure of 155 lb per sq in. The steam is used in four reciprocating engine generator units with a capacity of 1050 kw, and in winter the 5 lb exhaust steam is used for building heating. When Mr. C. H. Langs, the chief engineer, was called upon to provide air conditioning for the offices of a large tenant and a restaurant, he decided to utilize the same steam used for heating in the winter time for cooling during the summer.

Thus, the unique feature of the installation is the way the steam jet unit fits into the plant heat balance. In the ordinary plant, the exhaust steam from the engines would be wasted during the summer months. However, with this equipment, this usually wasted steam is used by the booster ejectors for refrigerating purposes, thereby providing refrigeration at low cost.

The water used by the surface condenser of the steam jet unit is cooled by a cooling tower on the roof of the building, so that its only cost is the cost of pumping due to the friction in the pipe line and surface condenser, plus the tower head. The water leaves the tower basin through a vertical downward pipe, then through the surface condenser, through the 865 gpm circulating water pump which is driven by a 45 hp steam turbine, and back up to the tower, so that there is a "u" bend of pipe from the tower downward and back up to the tower. It is necessary only to move the water through this pipe, plus the cooling tower head, plus the friction in the system.

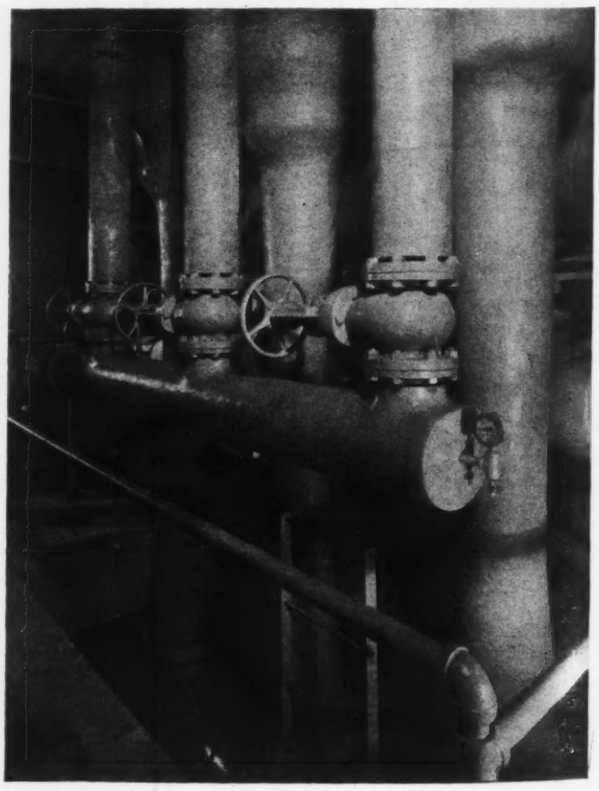
An Old Principle

Steam jet vacuum cooling is based on an old principle, literally "boiling the heat" out of the water, or other volatile liquid, used. To boil or evaporate liquids, a certain amount of heat is required. If this evaporation takes place in a closed vessel without adding external heat, the heat will come from the liquid itself. Since the boiling point of any liquid varies with the pressure, if the pressure in a closed vessel is reduced, the liquid, in order to come into equilibrium with the reduced pressure, flashes or boils off a certain amount of vapor, which process reduces the temperature of the remaining liquid to that corresponding to the pressure maintained.

Water at sea level (30 in. barometric pressure) boils at 212 F. On a high mountain this boiling point may be reduced to as low as 180 F, due to the reduction in atmospheric pressure. If the pressure in a closed vessel is reduced to, say, 0.3 in. of mercury, absolute, the boiling point of water becomes 45 F.

Due to the high latent heat of water, one pound of evaporation will reduce the temperature of 1000 lb of liquid approximately 1 F. In vacuum cooling, for every pound of water evaporated, there is removed from the remaining water about 1000 Btu. Thus, only a small percentage of liquid need be evaporated to accomplish a considerable amount of cooling. The vapor removed carries the heat away, and in the cycle of vacuum cooling

Fig. 1—Because of space limitations, the steam jet equipment was installed with the booster compressors vertical instead of horizontal, extending through the floor between the second and third basements. The flash chamber is at the top of this picture, and through the floor may be seen part of the surface condenser, a more complete view of which may be had in Fig. 2



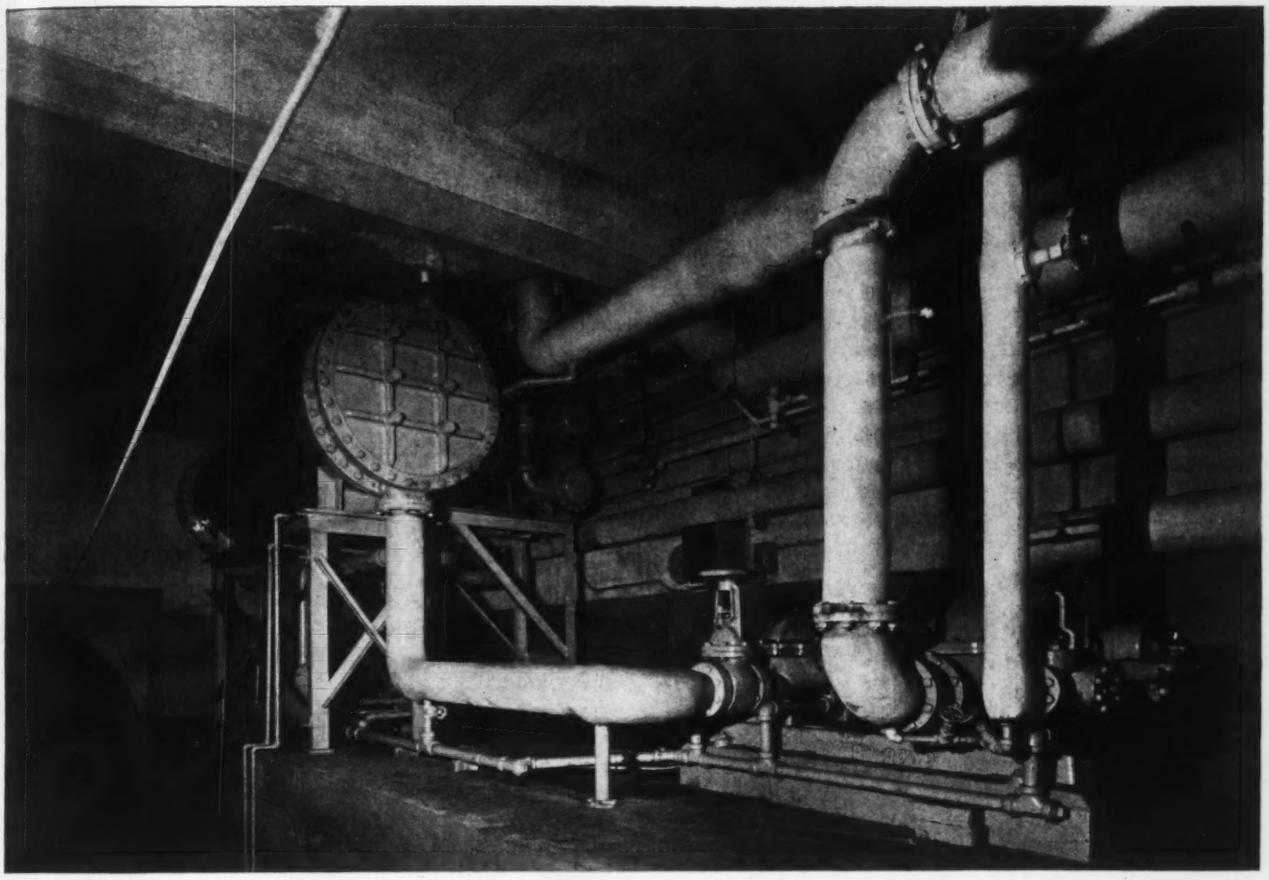


Fig. 2—This picture was taken in the basement below the one in which Fig. 1 was photographed. The surface condenser for the steam jet equipment is seen to the left and the turbine driven condenser circulating water pumping equipment at the right. Near the condenser is the secondary vacuum equipment, consisting of two stage ejector with surface type inter- and after-condenser. At the extreme left is the entrance to the freight tunnel system which serves buildings in Chicago's downtown area

this heat is absorbed by the cooling water in the booster condenser.

To obtain water cooled to any desired temperature above the freezing point, there is maintained in a vessel, called a flash chamber, a pressure (vacuum) equivalent to the temperature to which the water is to be cooled. Incoming warm water is sprayed into the flash chamber and chilled water is pumped out. The other equipment necessary is concerned only in maintaining the vacuum in the flash chamber and removing the flashed vapor.

Because of space limitations, the steam jet equipment at the Otis Building is installed with the booster compressors vertical and extends through the floor between the 2nd and 3rd basements. The total space required is 15x10x21 ft high.

The flash chamber or evaporator is 3 ft-6 in. inside diameter by 10 ft long, and is of heavy copper bearing steel plate. It is divided into three separate compartments, each of the three boosters connecting to a separate compartment and exhausting to a common surface condenser. It is necessary to have three separate compartments so that any one or two of the boosters can be shut off independently of the others in order to operate economically at reduced loads.

The surface condenser serving the three boosters condenses the booster nozzle steam, plus the flashed vapor, and is approximately 15 ft long by 3 ft in diameter. Each booster ejector is 16 in. in size, or, in other words, has a 16 in. suction. The surface condenser is made up of a copper bearing steel plate shell, with cast iron water boxes, and tubes of arsenical copper. The secondary vacuum, or air removal, equipment consists of a two-stage ejector, with surface type inter- and after-condensers.

Rated at 100 Tons

The equipment is rated at 100 tons of refrigeration, delivering 50 F chilled water, or, in other words, guaranteed to cool 480 gpm of water from 55 F to 50 F. Each of the three boosters is capable of handling one-third load.

The three boosters were all designed to operate on 5 lb gauge dry saturated steam pressure, while the secondary vacuum equipment, or two-stage ejector, to

When Chief Engineer Langs of Chicago's Otis Building was called upon to provide air conditioning service for offices and a restaurant, he decided to utilize the exhaust steam from his reciprocating engine driven generator units—used in winter for building heating—to do the job. A 100 ton steam jet refrigeration unit designed to operate on 5 lb steam was therefore installed, and 50 F chilled water is circulated to four spray type dehumidifier units

remove the air from the system, is operated on 125 lb steam pressure. It requires only a very small amount, and 5 lb pressure could not be used on it, as it must compress the air, plus saturated vapors from the condenser, to atmospheric pressure, which is not possible with 5 lb steam pressure. The surface condenser is designed to operate with 85 F condensing water from the forced draft cooling tower.

The steam turbine driving the circulating water pump is designed for steam at 125 lb pressure, and to exhaust at 5 lb gauge, so that it also supplies additional steam for operating the boosters if needed.

In general, with steam pressures of 100 lb gauge or more, a steam consumption of 25 to 30 lb per hr per ton of refrigeration may be assumed for such a unit, while for normal condensing water temperatures there will be required about 7 gpm of cooling water per ton of refrigeration. For low steam pressures, approximately 50 lb per hr of steam per ton of refrigeration might be assumed, while water quantities required would be approximately 10 gpm of cooling water per ton of refrigeration.

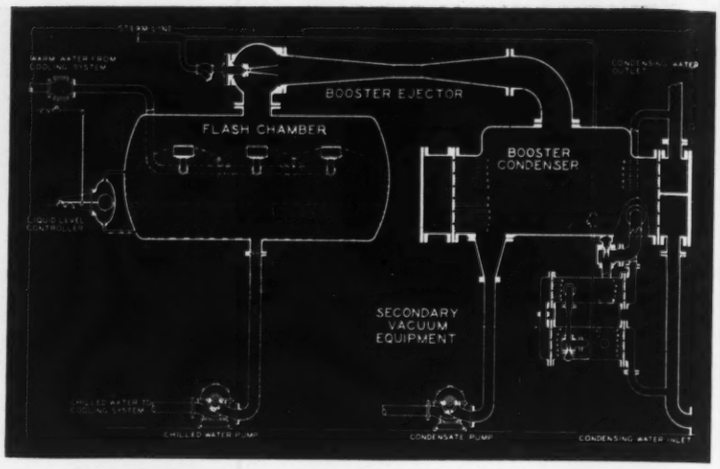
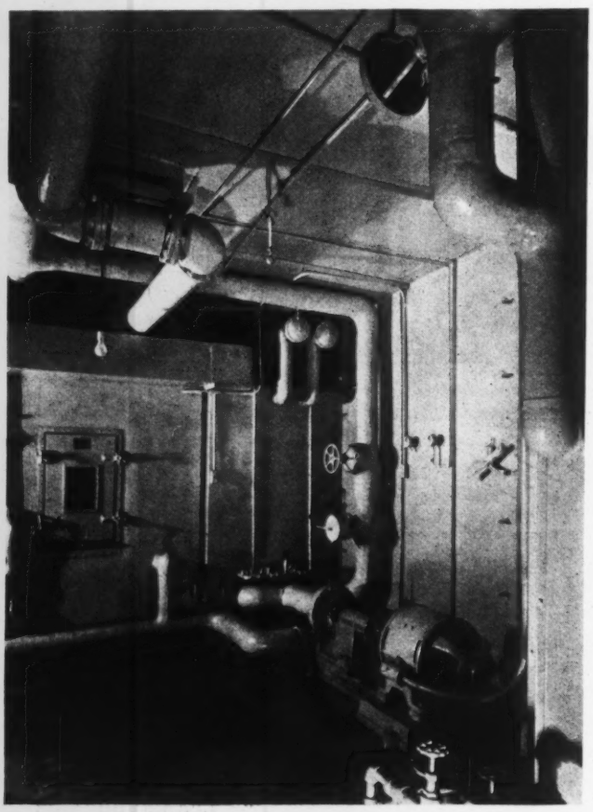


Fig. 4—Diagrammatic section through a typical vacuum cooling unit. At the Otis Building the booster ejectors or compressors are installed in a vertical rather than a horizontal position because of space requirements

Four Spray Type Dehumidifier Units

The 50 F water chilled by the steam jet machine is utilized in four separate spray dehumidifier units, two for the second floor office space, one for a board room on the second floor used by the same tenant, and the other for a restaurant on the first floor. A 480 gpm pump driven by a 20 hp motor supplies the water to these units. The second floor areas require 75 tons of the steam jet capacity and about 20 tons are required for the first floor restaurant, leaving 5 tons for the future air conditioning of a small shop.

Fig. 3—One of the four spray dehumidifier units supplied with chilled water by the steam jet refrigeration equipment



The office space is supplied from two identical duct systems and by two identical dehumidifiers, each system furnishing 9000 cfm of air to this space, and chilled water is circulated to these dehumidifiers by a 250 gpm pump driven by a 10 hp motor. The board room is supplied with 8000 cfm, and the dehumidifier is furnished chilled water by a 120 gpm pump driven by a 5 hp motor. The restaurant dehumidifier, installed in the sub-basement, handles 6000 cfm of air and has a 90 gpm water pump driven by a 3 hp motor. Each of the dehumidifier units has its own small circulating pump with the suction also connected to the chilled water supply and by means of automatic valves and dampers the proper temperatures and humidities are maintained in the conditioned areas.

Use was made of two existing ventilating systems in the general office space for air conditioning, but new risers were required and the duct outlets were changed because of partitioning to supply each individual office. Directional flow supply grilles were also installed. For the board room an entire new duct system with a dehumidifier in the basement was required.

An interesting feature of the installation is the booster unit installed in one of the private offices, occupied by an executive who is a hay fever victim. The supply duct comes up through the office in the pilaster and the booster unit comprises twin fans on a single shaft, with a specially treated air filter, installed under the window bulkhead, taking air from the supply riser, to insure the delivery of pollen free air.

The cooperation of Mr. C. H. Langs, chief engineer of the Otis Building, Mr. Charles E. Crone, Jr., of Wendt & Crone Co., the air conditioning contractors, and Mr. D. A. Johann, of the Elliott Co., in making the above information available, is gratefully acknowledged.

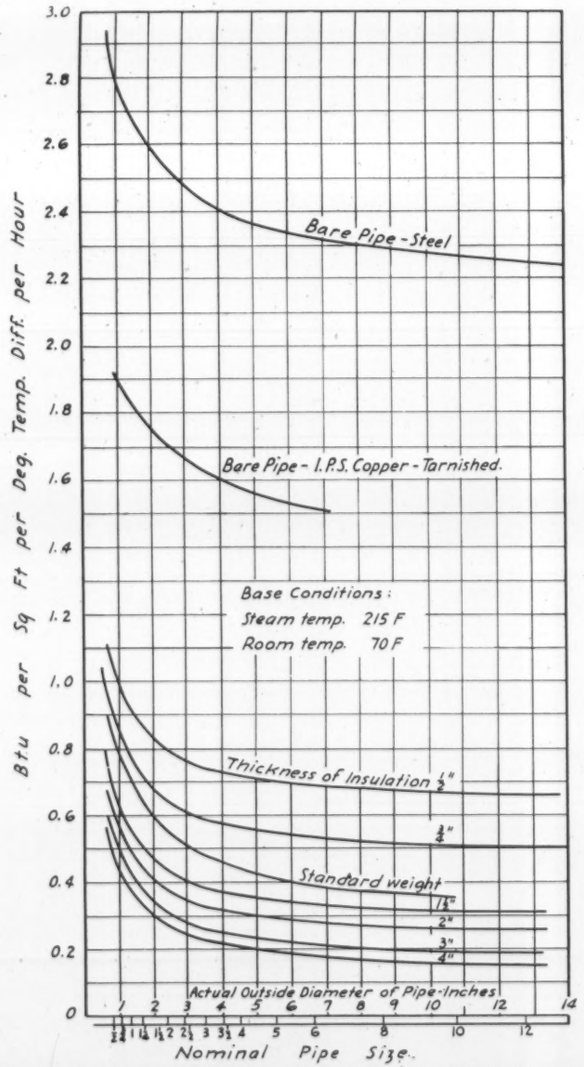


Fig. 1—Variation with Pipe Size of Rate of Heat Transfer Through Various Thicknesses of 85 Per Cent Magnesia Insulation and Through Bare Pipe (See paper by L. B. McMillan, *Trans. A. S. M. E.*, Vol. 48, p. 1277)

Selecting the Right Size Heating Boiler†

By Sabin Crocker*

Selection of the right size heating boiler is a vexing problem due to the number of factors involved and conflicting rating practices. Among the items affecting size selection are:

(1) Amount of connected (standing) radiation.
(2) Heat loss from bare and covered piping in supply and returns.
(3) Domestic hot water requirements.
(4) Satisfactory warming-up period from a cold or partially cold condition of the entire system.
(5) Efficiency with hard or soft coal, gas- or oil-firing as the case may be.
(6) Grate area with hand-fired coal, or fuel burning rate with stokers, oil or gas.
(7) Combustion space in the furnace.
(8) Type of heat liberation, whether continuous or intermittent or a combination of both.

All these factors have a definite bearing on the size boiler which should be selected for a given job, and should be taken into account in determining the proper amount of boiler heating surface and the requisite rate of heat liberation in the furnace.

Absorbing Capacity of Boiler Surface

Depending on the size of the job, the maximum heat absorbing capacity of the boiler surface should be from 1¼ to 2 or more times the heat emission rate corresponding to the summation of Items 1, 2 and 3 and the heat liberating rate in the furnace should be correspondingly greater, depending on the boiler efficiency; otherwise action will be sluggish. With an under-size steam boiler it even may be impossible to get up steam pressure from a cold start with forced firing, while with an under-size water boiler it may require several hours to raise the water temperature to the desired point.

Manufacturers' Ratings

Confusion has resulted from the practice of some manufacturers rating boilers as good for so many square feet of equivalent connected radiation (represented by the summation of Items 1, 2 and 3), while other manufacturers have based their ratings on heat absorption in the boiler which is equivalent to including Item 4 with Items 1, 2 and 3. Where boilers are bought on a competitive basis, there may be a tendency to skimp the size unless pains are taken to analyze the heating surface and fuel burning capacity of the boilers offered by different bidders. Even where competition is not a factor, it frequently happens that under-size boilers are installed through failure to appreciate the true significance of published ratings.

The bearing which the items listed above have on the selection of an adequate boiler for any job can be brought out to best advantage by taking them up in detail one at a time.

Item 1—Computation of Load Due to Connected (Standing) Radiation

The commonly accepted basis for computing radiation load is the nominal square foot of connected or standing steam radiation. Different types of radiation (column, wall, pipe coil, bare pipe, indirect, etc.) have differ-

*Engineer, The Detroit Edison Co. Member of Board of Consulting and Contributing Editors.

†Summarized and revised from a series of three articles published in Heating, Piping and Air Conditioning, February-March-April, 1932.

Here is the first installment of a new series of "summary sheets," on this and the preceding page. The purpose of these sheets is to make available for quick reference for everyday use information needed in heating, piping and air conditioning work. . . . For complete information, refer to the original article from which these data were obtained, as the purpose of these summaries is not to take the place of the article, but rather to put the essentials into convenient reference form. . . . The Editors will appreciate your suggesting what type of material you'd like presented in this way. . . . Address HEATING, PIPING AND AIR CONDITIONING, 6 North Michigan Avenue, Chicago, Ill.

ent heat emissions per square foot of actual surface. It is necessary, therefore, to reduce all surface to a common nominal basis, which for steam is accepted in the trade as one square foot of column (or tube) radiation containing steam at 215 F and in a room with an air temperature of 70 F. The heat emission of a nominal square foot of column steam radiation under the above conditions is defined as 240 Btu per hr, which corresponds to a condensation factor of approximately ¼ lb per hr. The heat emission of a square foot of hot water radiation depends, of course, on the average water temperature maintained, which varies to a considerable extent in different types of open and closed systems. A frequently used value for heat emission of hot water column radiation is 150 Btu per sq ft per hr, on which basis it takes 1.6 sq ft of hot water radiation to equal 1 sq ft of steam. This corresponds to an average water temperature of 170 F with a room temperature of 70 F. Values sometimes used for variations of these systems are 200-220 Btu per sq ft per hr for vapor system radiators and 200-240 for accelerated hot water radiators. This article will be confined in the main to steam heating with the understanding that the same general principles for sizing boilers apply to the others except where otherwise pointed out.

Table 1 offers a basis for reducing different types of radiation to a nominal basis of 1 sq ft of column radiation in air at 70 F.

In case the radiation contains steam or water at temperatures other than 215 or 170F, respectively, or the room temperature is other than 70 F, the conversion factors of Table 2 can be used as a basis for computing the equivalent nominal square feet. This table can be utilized in converting radiation of vapor or vacuum systems operating at temperatures other than 215 F into sq ft of 240 Btu radiation.

In the case of unit heaters, wall cabinets, and the like, the manufacturers' ratings should be used as a basis for computing the equivalent nominal radiation. After all installed radiation has been converted to its equivalent sq ft of nominal radiation, it should be summed up with the insulated and bare pipe losses and water heater load computed as will be described.

[To be continued.]

Table 1—Approximate Heat Emissions, Condensation Factors and Equivalent Ratios for Various Types of Radiating Surfaces with Room Temperature of 70 F[a]

Fluid	Radiation	Heat Emission Btu per Sq Ft per Hr	Equivalent Sq Ft Column or Tube Radiation		Sq Ft Equivalent to One Sq Ft Column Steam	Condensation Factor, Lb per Sq Ft per Hr
			Steam[b] 240 Btu	Hot Water 150 Btu		
Steam at 215 F..	Column or tube.	240	1.00	1.60	1.00	0.25
	Wall or pipe coil.	300	1.25	2.00	0.80	0.31
	Bare pipe.....	360	1.50	2.40	0.67	0.38
Water at 170 F..	Column or tube.	150	0.62	1.00	1.60	.....
	Wall or pipe coil.	190	0.80	1.27	1.26	.....
	Bare pipe.....	225	0.94	1.50	1.07	.....

[a] These are average values which do not take into account the actual number or height of columns, tubes or coils in a radiator nor whether the surface is horizontal or vertical. For more complete data, reference may be made to the *A. S. H. V. E. Guide*, particularly the 1928 edition in the case of column radiators.

[b] This column also represents heat emission relative to column steam radiation.

Table 2—Conversion Factors for Computing Under Given Conditions (A) Heat Emitted by Column or Tube Radiation, (B) Equivalent Nominal Radiation[a] [b]

Steam Pressure		Temp. of Steam or Water In F	Temperature of Air					
Vacuum In. Hg	Lb Absolute		80 F	75 F	70 F	65 F	60 F	50 F
22.4	3.72	150	0.388	0.424	0.462	0.499	0.538	0.617
20.3	4.74	160	0.462	0.499	0.538	0.577	0.617	0.697
17.8	5.99	170	0.538	0.577	0.617[d]	0.657	0.697	0.782
14.7	7.51	180	0.617	0.657	0.697	0.740	0.782	0.868
10 9	9.34	190	0.697	0.740	0.782	0.825	0.868	0.955
6.5	11.52	200	0.782	0.825	0.868	0.911	0.955	1.045
3.4	13.03	206	0.832	0.876	0.920	0.967	1.010	1.101
Lb Gage[c]								
0	14.7	212	0.873	0.928	0.974	1.017	1.063	1.126
1	15.6	215	0.911	0.955	1.000	1.045	1.091	1.183
2	16.9	219	0.947	0.991	1.036	1.081	1.128	1.220
3	17.9	222	0.973	1.018	1.063	1.109	1.155	1.249
4	18.9	225	1.000	1.045	1.091	1.137	1.183	1.277
5	19.6	227	1.018	1.063	1.109	1.155	1.202	1.296
6	20.8	230	1.045	1.091	1.137	1.183	1.230	1.325
8	22.8	235	1.091	1.137	1.183	1.230	1.277	1.373
10	25.0	240	1.137	1.183	1.230	1.277	1.325	1.421

[a] Based on conversion factor of *A. S. H. V. E. Guide* equal to $[(tr-t_a)/(215-70)]^{1.3}$ where tr = fluid temperature, t_a = air temperature.

[b] (Conversion factor) × (heat emission of conductor containing 215 F fluid in air at 70 F) = (Btu per hr heat emission of one sq ft of radiation under given conditions).

(Conversion factor) × (sq ft of conductor operating under given conditions) = (equivalent sq ft of same conductor containing 215 F fluid in air at 70 F).

[c] Approximate.

[d] This factor based on the heat emission ratio of nominal hot water radiation to nominal steam radiation would be 150 ÷ 240 = 0.625.

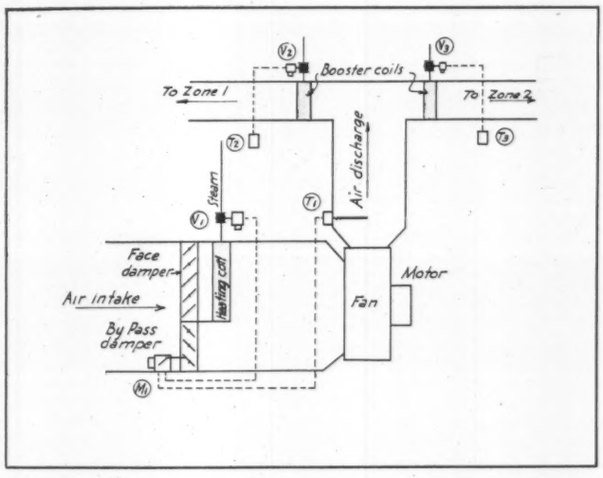
CONTROLS

By C. L. Ringquist*
Reference Sheet 10

Zone Control Systems

Zone control is not limited to any one type or design of system. The arrangements which follow have been selected at random, and embody either two or more of the various air conditioning functions together with zone control. There are, of course, numerous types of zone control systems which might be made up on each of the control systems as discussed in the preceding reference sheets. However, for illustration, the writer has selected typical arrangements which may be varied in accordance with the type of application or the wishes of the designer.

The following arrangement illustrates a control design which embodies heating, ventilating, and the zone control feature.



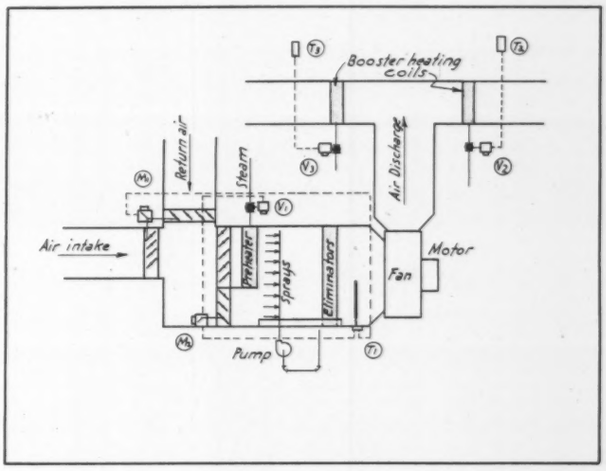
Cycle—In the branch duct to each zone there is a booster coil which takes care of the heating requirements for that particular zone. The air leaving the fan is kept at a constant discharge temperature at all times, and the necessary heat required by each zone is supplied by the booster coil.

Duct thermostat T_1 in the fan discharge maintains a constant temperature by varying the proportion of air passed through the heating coil and through the bypass damper. This is accomplished by operating modulating damper motor M_1, which controls the action of the face and bypass dampers. M_1 is equipped with an auxiliary switch connected to motorized or diaphragm valve V_1, of the on and off type, in the steam supply line to the heating coil. Whenever all of the air is being bypassed and the face dampers are closed, this auxiliary contact closes V_1, shutting off the steam to the coil.

In each zone there is a thermostat controlling a motorized or diaphragm modulating valve in the steam line to the booster coil. These are shown as T_2 for Zone 1, and T_3 for Zone 2; the valves in the steam line are noted as V_2 for Zone 1, and V_3 for Zone 2. These valves are of the modulating type, so as to supply just the right amount of steam to the booster coils, as required by the zone thermostats.

The following arrangement incorporates additional features: humidifying sprays, and an outside air intake damper and a return air damper for modulating the air supply in these intakes.

Cycle—Temperature controller T_1 acts to modulate damper M_1 to maintain a constant temperature leaving the sprays. M_1 is set for a minimum outside air quantity; that is, the outside air damper will close to a set minimum point, so that a definite

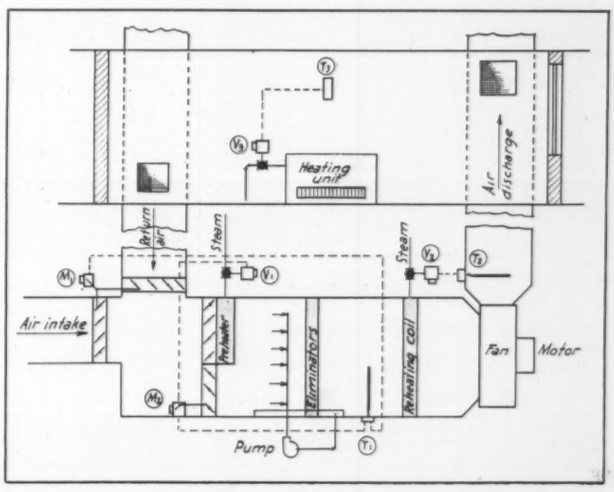


amount of outside air is being taken in at all times. When this set minimum quantity of outside air is being taken in and the temperature, as noted by T_1, continues to fall, the control switches from damper M_1 to damper M_2 and supplies heat to the mixture of air. Damper motor M_2 acts to modulate the amount of air passed through the preheater coil into the sprays, and the balance is passed through the bypass damper. M_2 is equipped with an auxiliary switch which controls the action of the two position valve V_1 in the steam supply line to the preheater coil. This auxiliary contact acts to close V_1 whenever all of the air is passed through the bypass damper.

Each of the zones has a thermostat, noted as T_2 and T_3, which controls a modulating valve (V_2 and V_3) in the steam supply line to the booster coil. These thermostats govern the amount of heat required in each of the zones.

Although only two zones are indicated, it is of course possible to have any number desired.

The following is a slightly different type of system than the two previously shown. The functions of this system are heating, ventilating, humidifying, and zone control. The heating is taken care of by a floor type unit heater situated in the space to be conditioned. Only one of the zones is indicated in the diagram for illustrative purposes.



*Air Conditioning Dept., The Trane Co.

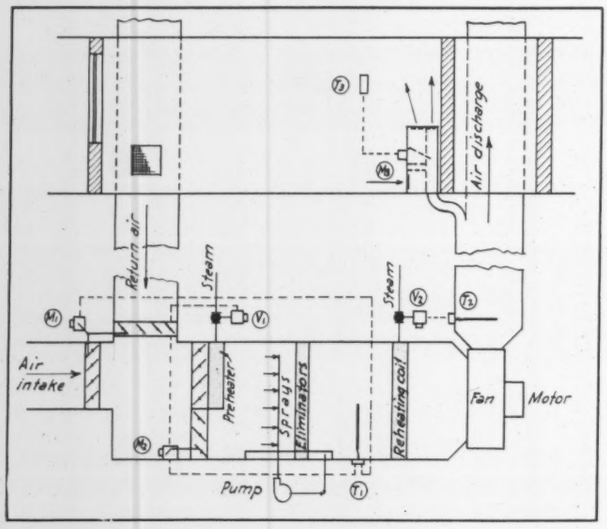
Cycle—Temperature controller T_1 maintains a constant air temperature leaving the sprays. T_1 is interconnected with damper M_1 which modulates the amount of outside air and return air to maintain a constant temperature at the location of T_1. M_1, which is of the modulating type, has a limit so that it cannot shut off the supply of outside air, but only modulate to a definite minimum closing position. When M_1 modulates to this position, the control is taken up from that point on by damper motor M_2, which operates to pass part of the air through the preheater and part through the bypass damper. M_2, also of the modulating type, is equipped with an auxiliary contact interconnected to two position motorized or diaphragm valve V_1 in the steam supply line to the preheater coil. Thus, when the face damper over the preheater coil is fully closed, this auxiliary contact closes off the steam supply by means of V_1.

Temperature controller T_2 in the fan discharge controls the amount of steam supplied to the reheating coil by means of modulating valve V_2.

A room thermostat (T_3) situated in each zone controls the amount of steam supplied to the heating unit through a modulating valve (V_3) in the steam supply line to this heating unit.

As explained, additional zones may be added to this system, but only one zone is used for purposes of illustration.

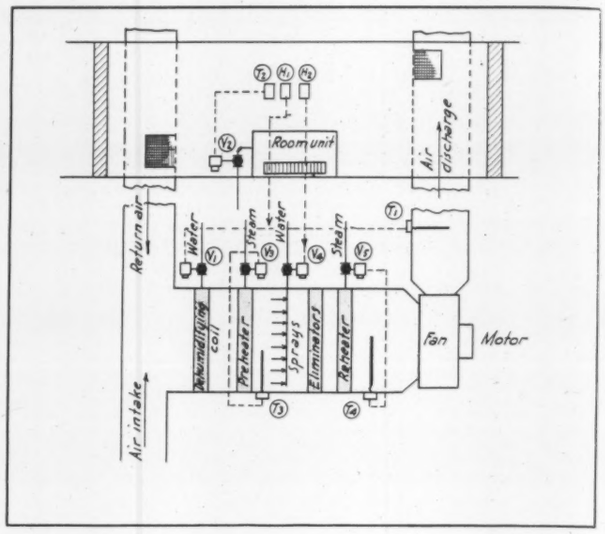
The following is a system very similar to the one just described, with the exception that there is no forced air circulation by the room units. The amount of air circulated in the room is by induced air current only. The air instead of being introduced through grilles is introduced in the back of the recirculating room unit. This air is discharged at a comparatively high velocity, and the inducing action of this air causes circulation of room air through the heating coil in the room unit.



Cycle—The control on the central system is exactly the same as outlined for the previous system. However, the controls on the room unit are slightly different, in that there is a room thermostat T_3 in each zone which operates a modulating damper motor M_3, which in turn operates to close or open a damper in the unit for more or less air circulation as required. M_3 is equipped with an auxiliary switch which operates a motorized or diaphragm valve in the steam supply line to the heating coil and closes this valve whenever the damper in the unit is entirely closed.

The following system incorporates cooling, dehumidifying, ventilating, humidifying, and zone control. This system is unique in that the room unit takes care of the heating requirements during the winter cycle and the sensible cooling requirements during the summer cycle.

Cycle—The system is designed so that the central dehumidifying system is not ample to take care of the cooling requirements



insofar as sensible heat is concerned. A higher temperature water or refrigerant is used in the room units, and these circulate the air for the summer cycle, removing the sensible heat from the air in the room.

The controls for the winter cycle comprise temperature controller T_3 in the central system after the preheating coil and before the spray chamber, which acts to modulate motorized or diaphragm valve V_3, thus controlling the amount of steam supplied to the preheater coil and maintaining the air at a constant temperature entering the sprays.

Room humidistat H_2, in the conditioned area, controls the action of modulating valve V_4 in the water line to the sprays, thus controlling the amount of water circulated to the sprays and the humidity.

Temperature controller T_4, in the central system between the reheater coil and the fan, controls the air leaving the reheater coil at a constant temperature by modulating valve V_5 in the steam supply line. Room thermostat T_2 in the conditioned area controls the action of motorized or diaphragm valve V_2, of the modulating type, in the supply line to the room unit for maintaining the temperature required in the conditioned area.

The controls for the summer cycle comprise duct thermostat T_1, in the discharge duct from the fan, controlling the operation of modulating motorized or diaphragm valve V_1 in the supply line to the dehumidifying coils. T_1 acts to maintain a constant temperature leaving the fan, and thus a constant humidity of the air discharged from the fan.

As an alternate to T_1, there is shown a room humidistat H_1, in the conditioned area, for controlling the action of V_1.

Sensible heat removal is taken care of by the room unit, and is controlled from thermostat T_2, which is of the summer-winter type. T_2 controls the action of modulating motorized or diaphragm valve V_2 in the refrigerant line to the room cooling unit. This room unit removes sensible heat only and does not take care of any dehumidifying.

One zone only is described here in order to illustrate the type of system.

A drop in accidents among workers in air conditioned factories was reported by Dr. Alexis A. Berestneff in an address before the Engineers Club of Hampton Roads, Va., because of "higher efficiency of human beings occupied either by mental or physical work."

"We fight epidemics and floods, we fight against wars because they cause damage, suffering and death," he said. "It must be realized that improper air conditions through inefficiency, accidents, disease and spoilage cause more damage, more suffering and more death than many other disastrous elements of nature. Air conditioning gives us the weapon to fight this disaster."

Heating · Piping
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AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

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Performance of Fin-Tube Units for Air Cooling and Dehumidifying

By G. L. Tuve* and C. A. McKeeman** (*MEMBERS*)
Cleveland, Ohio

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS *in cooperation with the Case School of Applied Science*

THIS paper undertakes to summarize and to compare several of the methods which are at present used for expressing the cooling and dehumidifying capacities of surface coils. Methods are suggested for making rating tests of any coil, or for selecting a coil for a given service when such ratings have been made, and the accuracy of these methods is discussed.

Test results indicate that many of the simple methods of calculating surface-coil ratings are only approximations. It is also demonstrated however that accurate ratings call for much experimental work of a very precise nature, and for involved calculations to interpret the results. It is because this accuracy requires more time than the practicing engineer or the manufacturer can devote to the problem, that a program of cooperative research along these lines has been undertaken.

Complexity of Problem

Attention has been called in the previous reports on this research[1] to the large number of variables affecting the performance of a fin-tube coil. A list of these variables for an air-cooling and dehumidifying coil is given in Table 1. All of the variables indicated in this table have been investigated to a greater or less degree, in this research project to date. In order to simplify the problem, however, the data and methods given in this paper have been applied only to copper fin-coils with cross-flow or mixed flow. (No true counterflow or true parallel-flow coils have been tested, and few such coils exist in practice.)

Many scientific papers are available on the various phases of the subject of extended-surface heat transfer, and a number of the laws, by which the variables of Table 1 are related, have been developed. It is recognized, for instance, that the heat transfer in an air-cooling coil may conveniently be divided into three steps: (1) Heat flow from air to outside metal surface; (2) Heat flow through the metal; (3) Heat flow from the inside metal surface to the refrigerant. While these three steps may be related by writing the usual equation for the overall coefficient of heat transfer in terms of the individual coefficients, this relation has not been extensively used in practical work, because the individual coefficients

Table 1—Variables Affecting Fin-Coil Performance

Coil

Surface-ratio: Ratio of air-side surface area to refrigerant-side surface area.

Type of fins: Round, square or continuous.

Shape of fins: Plain or crimped, ribbon or wedge-shape.

Fin bond: Integral, dipped, expanded, crimped, etc.

Material: Copper, aluminum, steel, cast-iron, brass.

Depth and piping: Depth of fins, number of tube-rows, tube spacing, counterflow, parallel flow, cross flow, mixed flow, etc.

Air

Velocity: Mass velocity through free area, or linear velocity through face area.

Temperatures: Dry-bulb, wet-bulb and dew-point.

Turbulence: Reynolds number, local eddies, stratification and distribution.

Refrigerant

Type: Liquid or direct-expansion vapor.

Velocity: Velocity of liquid, gas or vapor in tube.

Turbulence: Produced by velocity or by mechanical means, such as spirals or corrugated walls.

are not well known and are very difficult to determine.

Goodman[2] has recently outlined a promising method for finding both the air-side and the refrigerant-side coefficients. Pownall[3] and others have given a few data on the refrigerant-side coefficients, and Wells[4] has outlined a method for finding the *effective air-side coefficient*. These investigators have, in turn, built upon the earlier work of Knaus, Scanlan, Schmidt and others.[5, 6, 7, 8, 9, 10, 11, 12, 13] As a final result a number of conclusions have become fairly well established and are supported both by analysis and by experimental evidence. Some of these are given in Table 2. Certain of these statements are approximations, but they are much closer approximations than most of the assumptions given in Table 3.

Practical Solutions

Three methods are available for expressing quantitatively the performance of engineering equipment: (1) Performance tables; (2) Performance charts or graphs; (3) Performance equations.

But, as the number of variables increases above two or three, any of these methods becomes complex or cumbersome. Hence it has been the practice of fin-coil users to resort to various approximations which simplify the presentation of coil data. In Table 3 are listed a number of these common approximations.

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[1]For references, see Bibliography.

For presentation at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1937.

[2,3,4,5,6,7,8,9,10,11,12,13]For references see Bibliography.

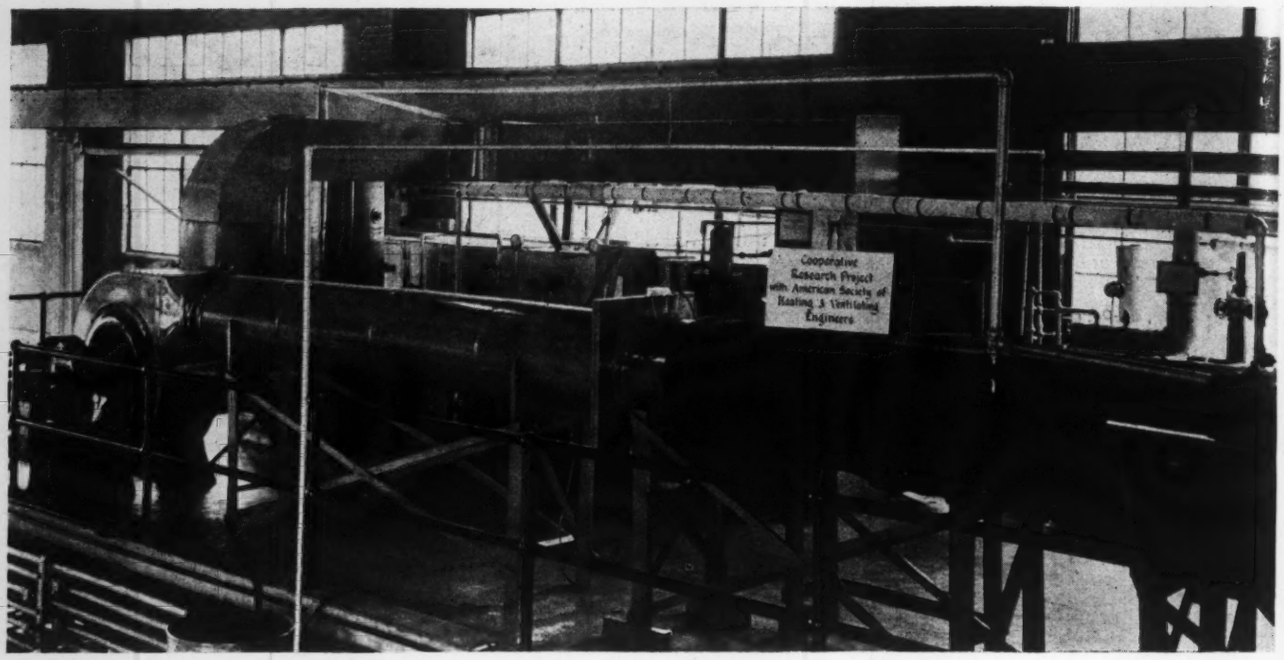

Fig. 1—General view of fin-coil test unit

In the following sections the results of a number of tests on cooling coils will be presented, and these will be correlated by showing how the test results agree or disagree with the statements given in Tables 2 and 3. Before presenting these correlations, however, it will be necessary to indicate the specific problems which the engineer is most frequently required to solve.

Table 2—General Statements Relative to Fin-Coil Performance

Item No.	Statement	References		
		List No.[a]	Author	Page
1	Over a limited velocity range, the overall heat - transfer coefficient varies with a fractional power of the air velocity, i.e., $U = C V^n$.	10 2 1–*a* 1–*b*	McAdams Goodman Tuve-McKeeman Tuve	266 240-41 427 589
2	The heat-load ratio may be determined from the slope of a straight line on the psychrometric chart which connects the points representing the inlet and the outlet conditions.	9 3 8 4	Knaus Pownall Goodman Wells	23–82 211 90 665
3	The relations between dry-coil capacity and dehumidifying capacity may be expressed in terms of the characteristics of this straight line drawn on the psychrometric chart through the entering and the leaving air conditions.	6 4 9	King-Knaus Wells Knaus (See also Fig. 6 herewith)	283 665 25–82
4	With all other conditions constant, the total load varies with the wet-bulb temperature of the entering air, irrespective of dry-bulb or dew-point temperatures.	13 4	Scanlan Wells	197 665
5	The effective surface temperature corresponds to that dew-point temperature of the entering air at which condensation appears on or disappears from the air-side surface.	4	Wells	667
6	The difference between the effective surface temperature and the refrigerant temperature varies directly as the total load, irrespective of the nature of the load.	3 4	Pownall Wells (See also Fig. 3 herewith)	211 667
7	The heat transfer coefficient is increased by increasing air turbulence or local eddies in the approaching air stream.	1–*a* 6	Tuve-McKeeman King-Knaus	427 283
8	Separate tables may be prepared and used for the sensible-heat and the latent-heat transfer capacities of a coil.	1–*b*	Tuve (See also Fig. 7 herewith)	594

[a] See Bibliography.

Coil Ratings and Coil Selections

Three typical engineering problems will be considered:

(1) How should a coil be rated (i. e., its performance determined by a standard method), and how may the performance be expressed in order that one coil may be compared with another?

(2) If the coil performance under certain conditions is known, how may the performance under another set of conditions be determined?

(3) Given the outdoor and indoor air conditions, and the calculated sensible and latent heat loads, how may a coil be selected which will handle these loads and produce the desired conditions?

There are at present almost as many methods for rating a coil as there are coil types or designs. The American Society of Heating and Ventilating Engineers might well look forward to the establishing of a Standard Rating Code, similar to those it has adopted for radiators, convectors and boilers. It is not too early for members to express their views on this subject, because much discussion will be needed in order to arrive at the best methods.

The problem of determining the changes in coil performance when the conditions are changed is only partially solved at present. If the change is merely one of air condition, refrigerant temperature, or air or refrigerant velocity, a close approximation to the new performance can be obtained. But for changes in coil design, in coil depth, in type of refrigerant, or in the type of flow as when changing from cross-flow to counter-flow, more information is needed and attention is called to the Program of Research outlined hereinafter.

The solution to the problem of selecting a coil for a given set of conditions will depend on the nature and

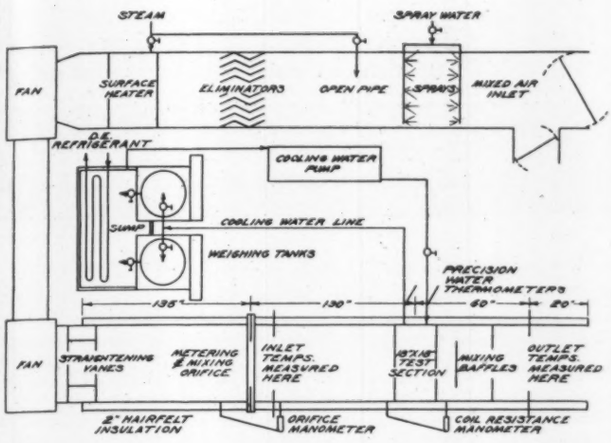

Fig. 2—Line diagram of apparatus shown in Fig. 1

the completeness of the rating-data available on the coil-type being considered. Many selections require compromises of one kind or another, and it must not be expected that when the air conditions and the loads are fixed, the refrigerant temperature, air velocity, space requirements and similar factors may also be arbitrarily prescribed. This amounts to *telling the coil what to do,* rather than making a reasonable selection and then calculating what that coil *will* do. It is a simple matter to write specifications which cannot be met by surface coils at all. Special arrangements such as by-passing, reheating, etc., will enable surface coils to meet certain of these difficult requirements.

Table 3—Common Approximations Relative to Fin-Coil Performance

Item No.	Assumption[a]	For Checking[b] Accuracy See:
1	A change in the number of rows of tubes (coil depth) does not change the heat-transfer coefficient.	Ref. 1*b*, p. 592
2	A simple factor may be used to correct the heat transfer coefficient for the rate-of-flow of the refrigerant.	Ref. 1*b*, p. 593
3	The coefficient of sensible heat transfer is unaffected by the magnitude (or absence) of accompanying latent heat transfer.	Fig. 6
4	A simple correction factor may be used for wet-coil operation, dependent on temperature difference between entering dew-point and refrigerant.	
5	Air always leaves a fin-coil in the saturated state.	Fig. 8
6	Air always leaves a fin-coil at a fixed relative humidity, irrespective of operating conditions.	Fig. 8
7	When a straight line is drawn on the psychrometric chart from entrance to exit air conditions, it will intersect the saturation curve at the effective surface temperature.	Ref. 4, p. 665 Ref. 5, p. 299 Fig. 3
8	A straight line on the psychrometric chart from entrance to exit air conditions represents also the succession of state-points, i. e., the condition curve for the air.	Ref. 2, pp. 228 and 250
9	The difference between effective surface temperature and refrigerant temperature is constant.	Ref. 4, p. 668 Fig. 3
10	The effective surface temperature is equal to the mean between the wet-bulb temperature of the exit air and the temperature of entering refrigerant.	

[a] Some of the assumptions are incomplete statements, but it is assumed that the reader will recognize the omissions.
[b] For references, see Bibliography.

Results of Tests on Typical Fin-Coils

The tests reported in the earlier papers on this project[1] were made with a draw-through arrangement taking air from the room, but the equipment has now been rearranged for blow-through operation, with air supplied from a conditioner. Fig. 1 is a general view of the equipment, with the testing tunnel in the foreground and the air conditioner in the background. The diagram of Fig. 2 shows the relative arrangement of parts, with principal dimensions.

Results are given herewith on two of the four types or makes of coils which have been tested to date in this research. Results on the other two coils lie in the intermediate region, and will be presented in a later report. *Coil No. 1* was a 4-row thick-fin, round type, with a low ratio of fin surface to tube surface, and with a high-turbulence water path. *Coil No. 2* was a 4-row continuous-fin coil with thin fins, a large ratio of extended surface to tube surface, and a smooth-tube water path. Most commercial coils with 4 rows of tubes will therefore lie within the region covered by these two coils.

Fig. 3 shows the temperature difference (or temperature gradient), between the refrigerant temperature and the effective surface temperature as defined in Table 3, item 7, and in Table 2, item 6.

Fig. 4 shows the dry-coil performance of these units over a range of face velocities from about 300 fpm to 700 fpm, and with water velocities in the tubes about 1 ft per second. The overall coefficient is the heat transfer in Btu per square foot of surface area, per degree log mean temperature difference between air and refrigerant.

Fig. 5 gives the factor by which the dry-coil coefficient from Fig. 4 must be multiplied in order to obtain the total heat transfer coefficient with dehumidification. This factor is plotted against the load-ratio, total-heat load to sensible-heat load.

Fig. 6 shows several other methods of plotting the same data as that shown by Fig. 5. Some engineers wish to express the load-ratio in terms of sensible to total, or of sensible to latent, and the corresponding wet-coil factors are given in Fig. 6. If an engineer prefers to select coils on the basis of sensible heat only (instead of total heat transfer), the factors in the lower row of graphs should be used. All the curves of Fig. 6 give identical results, and they merely represent different methods of plotting the data of Fig. 5.

Fig. 7 gives the heat-transfer coefficients for latent heat only. Since the heat potential for latent heat transfer is the difference between dew-point (i. e., condensing temperature), and refrigerant temperature, the mean temperature difference for the latent-heat coefficient must be calculated on the basis of the dew-point temperatures.

Fig. 8 shows the difference between dry-bulb and wet-bulb temperatures at exit, and the approximate relative humidity of the exit air, for 4-row coils of usual design, based on the same two coils, and covering the range of heat-load ratios.

Discussion of Data in Figs. 3 to 8

In order to indicate the care with which the data here presented have been obtained, a number of the precau-

[1] For references, see Bibliography.

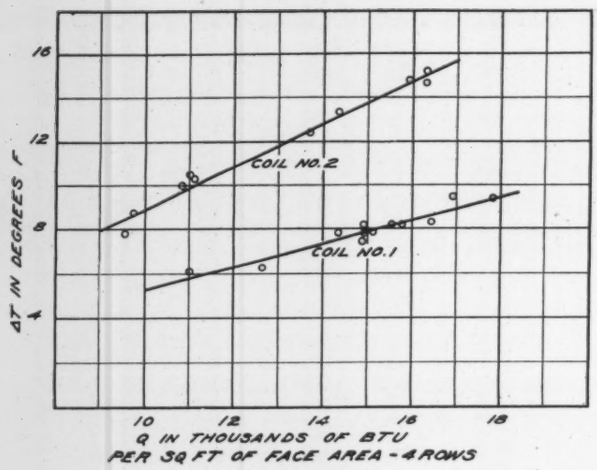

Fig. 3—Temperature gradient ΔT between refrigerant and air-side surface

tions in test procedure will be enumerated:

1. Water temperatures were read to 0.01 F, on accurately matched thermometers.
2. Outlet water was mixed by three 90 deg elbows before reaching the thermometers.
3. Air temperatures were read to 0.1 F on four equally-spaced thermometers at entrance, and four at exit.
4. Exit air was mixed by two orifices in series, with a target between.
5. Wet bulb thermometers were continuously supplied with water through long precooling wicks, with capillarity assisted by gravity.
6. All temperatures were controlled within 0.3 F during each 60-min run.
7. Air quantities by heat balance must check air measurements by orifice within 5 per cent.
8. Dehumidification by heat balance must check dehumidification by weight of drip-water within 5 per cent.
9. A complete set of readings was taken every 4 min for each 60-min run.
10. A preliminary equalizing period of at least one hour preceded each run.

On Figs. 5, 7, and 8 the approximate air velocities for the various tests are indicated as follows: squares indicate velocities below 300 fpm; circles indicate velocities around 400 fpm; triangles indicate velocities above 500 fpm.

It was found to be entirely possible to make repeat or duplicate runs on dehumidification, with all conditions steady and all check-readings satisfactory, and yet have the points separate as much as is indicated by Fig. 5. There is no doubt about an actual difference of performance between such runs. A similar difficulty has been reported by investigators in all fields dealing with surface condensation. Great differences have been shown to exist in the rate of heat transfer in condensation, depending on whether the condensation is being formed as *film condensation* or as *drop condensation*. It is the opinion of the authors that similar differences exist in the case of dehumidifying coils, depending perhaps on the condition of the surfaces, and their immediate-past history. Differences in the nature or completeness of condensate drainage from the fins also affect the performance. It is, therefore, not to be expected that ratings of dehumidifying coils can be set within the same tolerance limits as those used for dry-cooling coils.

Application to Practical Work

From data such as those plotted in Figs. 3 to 8, various methods of procedure are available for obtaining answers to the three questions already listed under Coil Ratings and Selections.

As a typical problem, consider the case of a fixed condition of the air entering the coil, and specified sensible and latent heat loads to be carried. This fixes a starting point on the psychrometric chart, and fixes the slope or direction of the line on this chart which connects the air conditions at entrance and exit.

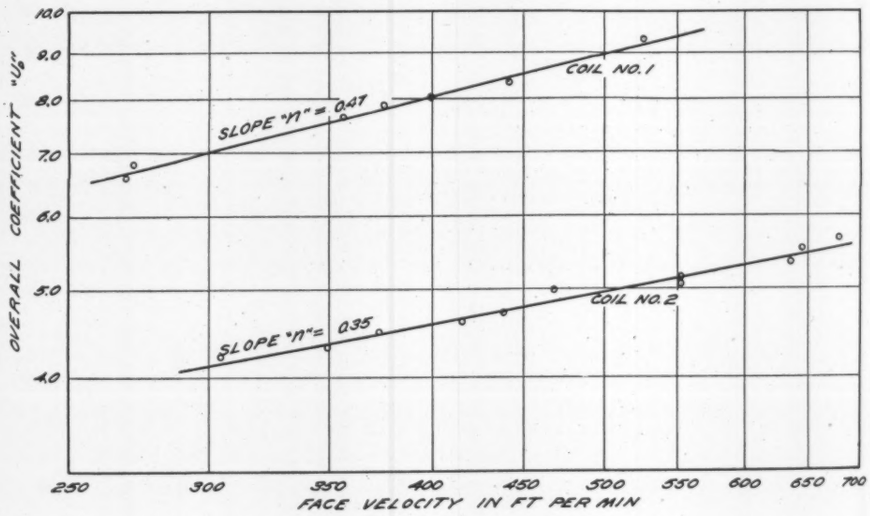

Fig. 4 — Overall coefficients of dry coils per square foot of air-side surface

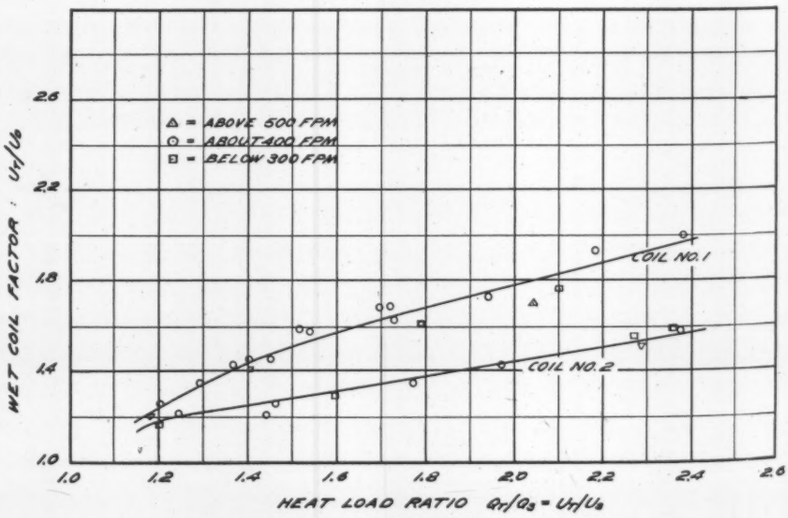

Fig. 5—Typical wet-coil factors for total heat

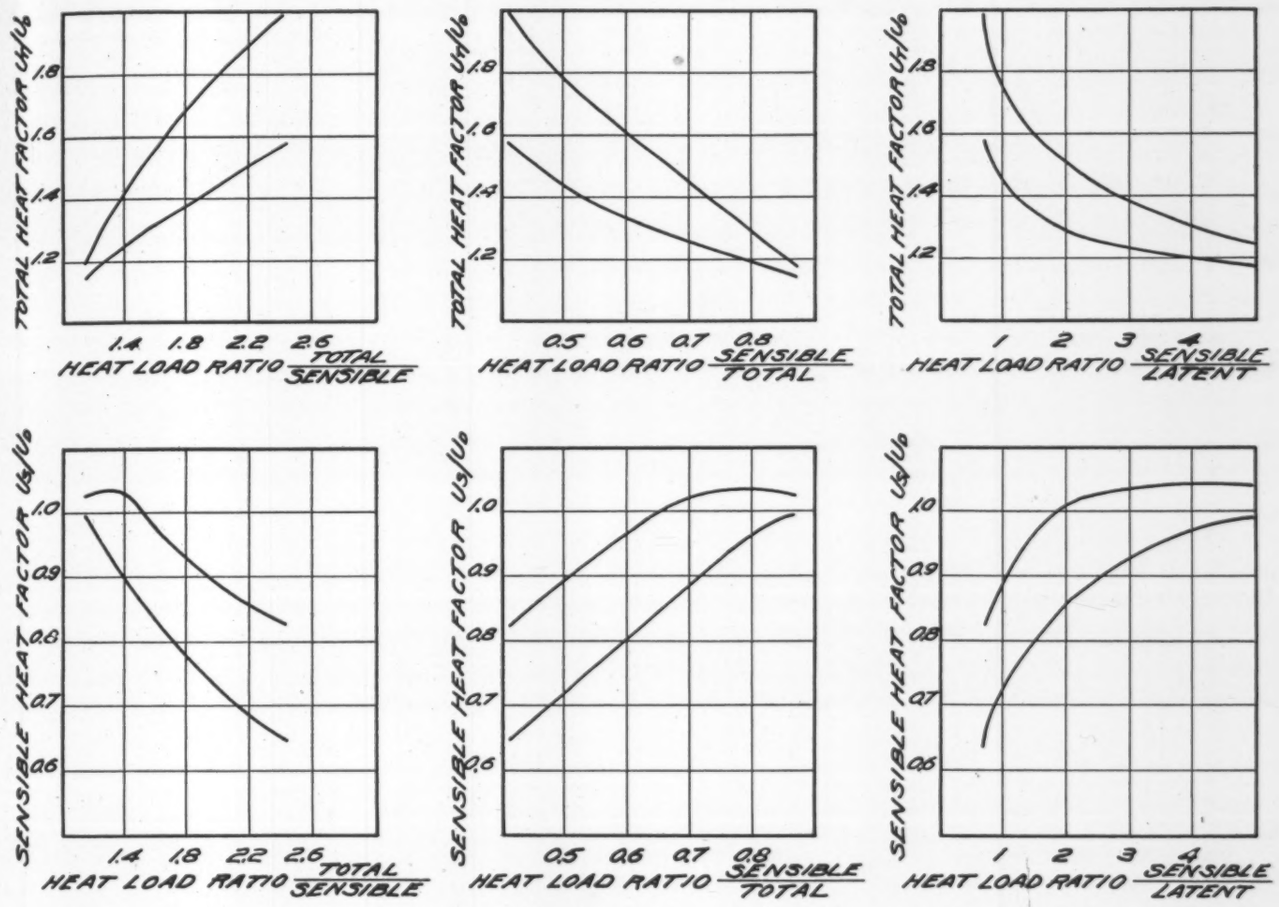

Fig. 6—Six methods of plotting the wet-coil factors

It is then necessary to select the type of coil to be used, and the kind and method of application of the refrigerant. Assume, for example, that a style of unit such as the No. 1 coil in Figs. 3 to 8 is to be used, with water as the refrigerant, and a small water-temperature rise.

The remaining variables are: (1) Coil depth; (2) Refrigerant temperature; (3) Air velocity; (4) Face area; (5) Exit air conditions. In the present example, assume that a 4-row coil is to be used (since the graphs herewith are based on 4-row coils). The problem may then be solved either on the basis of a fixed mean refrigerant temperature, or on the basis of a fixed air velocity. In either case the exit air conditions and the face area will depend on the properties of the coil, and hence these conditions cannot be arbitrarily fixed.

Summarizing, the conditions for this example are:

Given: Total load; load ratio; entering air conditions; air velocity (or refrigerant temperature).

Required: Exit air conditions; refrigerant temperature (or air velocity); coil face area.

For a specific solution a face velocity of 450 fpm will be assumed, with a total load of 60,000 Btu per hour, of which two-thirds is sensible heat and one-third latent heat, and air entering at 85 deg dry-bulb, 50 per cent relative humidity. Procedure by the various methods is as given:

Step 1. Locate the initial condition point on a non-logarithmic psychrometric chart. Draw a straight line through this point for a total-to-sensible heat-ratio of 1.5, at the proper angle for the particular chart used. This line may be either drawn through a second point which has been calculated from the psychrometric tables, or the table of angles supplied with the particular chart may be used.

Step 2. Using Fig. 8, determine the exit air conditions for $Q_T/Q_S = 1.5$. These are: Dry-bulb wet-bulb difference = 3.5 deg, relative humidity approximately 82.5 per cent. Referring to the psychrometric chart, the exit conditions for this coil are found to be: Dry-bulb = 64.7 deg, wet-bulb = 61.2 deg. From the wet-bulb temperatures at entrance and at exit, the heat extracted from each pound of dry air is found to be 7.6 Btu.

Step 3. The load per square foot of face area may now be calculated by the heat-balance equation:

$$Q = \frac{\text{face velocity feet per minute} \times 60 \times 0.075}{\text{pound mixture per pound dry air}} \times \text{heat extracted per pound}$$

= 15,220 Btu per hour per square foot of face.

$$\textit{Then}: \text{Face area} = \frac{60{,}000}{15{,}200} = 3.95 \text{ sq ft.}$$

Step 4. To determine the refrigerant temperature:

1. *Surface Temperature Method.* From Fig. 3, at this load, the difference between surface temperature and refrigerant temperature is 8.1 deg. The surface temperature (defined in Table 3, Item 7) from the psychrometric chart is 57.5 deg hence the refrigerant temperature is 49.4 deg.

2. *Overall Coefficient Method.* From Fig. 4 the dry-coil coefficient is 8.55. The wet-coil factor, total heat basis, from Fig. 5 is 1.51, hence the overall coefficient for total heat, $U_T = 12.9$. Solving for the logarithmic mean temperature difference (dry-bulb basis), using 15,200 Btu per square foot of face and 51.7

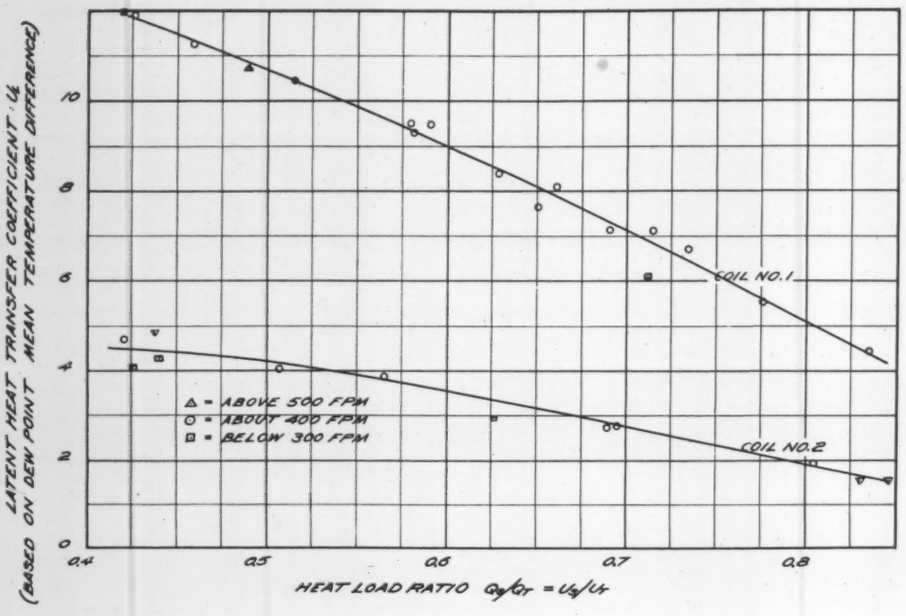

Fig. 7—Latent-heat coefficients based on dew-point temperature difference

sq ft of surface per square foot of face (as measured from this coil), the result is: mtd = 23.75 deg. Solving for refrigerant temperature (by formula, tables or chart), the result is 49.8 deg by this method.

3. *Latent Heat Method.* From Fig. 7, the latent heat coefficient U_L = 7.75 (based on dew-point temperatures). Solving for dew-point log mean temperature difference, the result is: mtd = 12.6 deg. The refrigerant temperature by this method is then found to be 49.5 deg.

While the example given uses the same known conditions to illustrate all three methods of solution, each method is best applied within a certain range. The selection of method depends upon which conditions are known and which are unknown.

Conclusions and Program of Research

The methods of coil selection illustrated by the examples are free from the cut-and-try procedure which characterizes several of the methods now in common use. The methods here presented also give definite solutions in accordance with the characteristics of the coil, instead of *telling the coil what to do*. It is a common fault to specify too many conditions and to expect a given type of coil to satisfy all of them. The above methods offer check solutions by which this common error may be detected.

It should be emphasized that all of these methods are subject to discrepancies arising from inaccurate construction on the psychrometric charts, and to disagreements between the various charts themselves. It is particularly easy to make an error of one-half degree or more in the surface temperature, where that method is used.

More data should of course be accumulated as rapidly as possible, to include a variety of coil types and of operating conditions. It is probable that after this has been done, a coil may be completely rated by merely making a few dry-cooling tests as shown in Fig. 4, because there is apparently a close correlation between Fig. 4 and the Figs. 3, 5, 6, 7 and 8. Data on this correlation are being prepared for a later report. These methods therefore have considerable promise as bases for Test Code procedure.

The main purpose of the research program in the immediate future is to refine the accuracy and to increase the comprehensiveness of the results obtained from the methods outlined in this paper. Data on other depths of coils are necessary, as well as data on other coil designs and operating conditions. Some additional results have already been obtained and are being completed as rapidly as possible.

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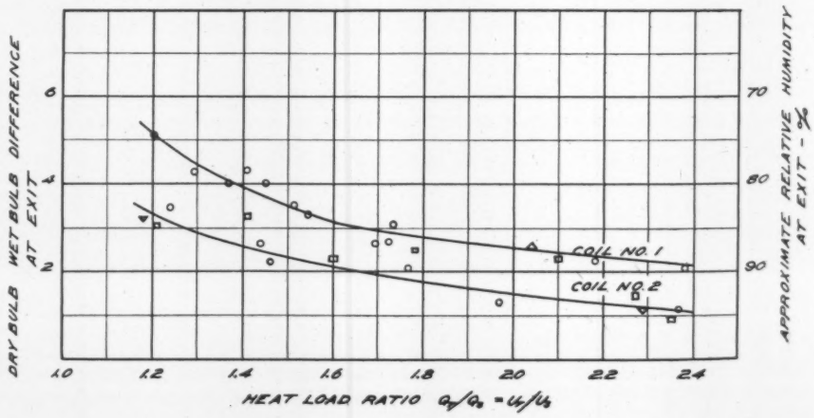

Fig. 8—Air condition at coil exit

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Correlating Thermal Research

As a part of the efforts of the A. S. H. V. E. Research Laboratory to correlate research in thermal engineering carried on by the many institutions engaged in such work, and to disseminate the published results of such studies together with other reports of progress in this field, in order to make them available to the membership of the Society, there will be published monthly on this page a limited number of brief abstracts of articles which it is believed will be of greatest interest to all concerned. For more complete lists address The Director, A.S.H.V.E. Research Laboratory, U. S. Bureau of Mines Experiment Station, Pittsburgh, Pa.

F. C. Houghten, *Director*. W. A. Danielson, *Chairman*.

COMMITTEE ON RESEARCH

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A. S. H. V. E. RESEARCH

A Page of News Notes and Comment

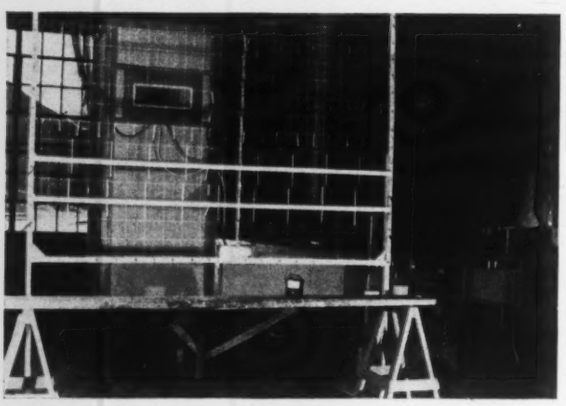
By Brewster S. Beach*

SPECIAL EQUIPMENT AT CASE AIDS AIR DISTRIBUTION STUDY

G. L. Tuve

With the importance of scientific air distribution being stressed in current air conditioning research, interest in this problem is centered on the work being carried on at the Case School of Applied Science, Cleveland. G. L. Tuve, Professor of Heat-Power Engineering, has been in consultation with the A. S. H. V. E. Laboratory at Pittsburgh respecting the experimental work that is now in progress. The photograph published herewith shows the nature of the set-up in the Case School laboratory. It consists of an F-12 air cooling unit and a vertical plenum air supply chamber with grille mounting at a height of 10 ft from the floor.

"The equipment," Professor Tuve explains, "will accommodate any size and type of grille up to 12 in. x 30 in. face. By means of portable guide frames, divided into 6 in. squares, we make velocity and temperature traverses at various distances in front of the grille. Ribbon indicators are mounted in these squares to show the approximate contour of the stream. We have already done some work on various grille styles, and we think that it may be possible to develop a set of equations relating the variables of face velocity, maximum length of throw or blow, drop of the stream and volume or size of the stream-envelope."



Experimental apparatus, Case School of Applied Science, for testing velocity and temperature of entering air within a room

Case School experimenters hope to develop data for a straight flow grille of the low resistance type as the basis for further performance studies of other types in relation to the straight-flow unit. The activity is in general charge of Professor Tuve who is a member of the A. S. H. V. E. Committee on Research and member of the Committee on Air Distribution which includes Ernest Szekely, *chairman*, S. H. Downs, M. K. Fahnestock, F. J. Kurth, D. W. Nelson, C. H. Randolph, J. E. Shoen and J. H. Van Alsburg.

*Director, Information Service, Committee on Research.

INSULATION RESEARCH INTENSIFIED IN COLLEGE LABORATORIES

F. C. Houghten

Recently returned from a trip to several cooperating universities, F. C. Houghten, director of the A. S. H. V. E. Research Laboratory, reports unusual interest in the study of the newer problems of insulation, particularly the matter of condensation of moisture on or within the insulation substance as the result of what is called "breathing."

Primary interest, Mr. Houghten explains, centers today not in the conductivity or heat resistance properties of insulating materials as they may be manufactured, but rather in the performance of the materials in practice as they may be affected by condensation within the walls themselves.

Mr. Houghten said that at the University of Minnesota, facilities are being provided for maintaining under control, severe winter conditions in a large space in which a number of insulating materials will be tested. At the University of Wisconsin keen interest, he reported, is being displayed in the improvement of the rural home with the problem of insulation being an important part of their studies.

At Purdue University research, that fits closely with the aims of the A. S. H. V. E. Committee on Insulation and which makes possible cooperative work between them, is being carried on in the application of insulation and its performance during its life afterward. The major part of this study centers around the economic amount of insulation to be used, the type to be used in order to give the desired performance in later years, as well as the installation features.

NEW RESEARCH REPRESENTATIVE

Dexter Purinton, member of the New York Chapter, is the latest addition to the list of Research Representatives who are being appointed by the Society's local chapters in accordance with a plan described in the May issue of the JOURNAL. Mr. Purinton is associated with the Mahony-Troast Construction Company, New York, and for many years was in charge of the mechanical engineering department of Voorhees, Gmelin & Walker, New York architects. There are now eleven research representatives serving as local contacts between the general Committee on Research and the local chapters.

RESEARCH PUBLICITY

The circulation of magazines, newspapers, business periodicals and trade journals, in which educational news of A. S. H. V. E. Research has been appearing since the beginning of the year under the public relations program to broaden public knowledge of the Society's research activity, now totals several million. *Forbes* and the *Scientific American* are among the general magazines now working on material for articles supplied them through the Society, which are scheduled for publication during the summer.

Corrosion Studies in Steam Heating Systems

By R. R. Seeber* (*MEMBER*) and Margaret R. Holley** (*NON-MEMBER*)
Houghton, Mich.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS *in cooperation with the Michigan College of Mining and Technology.*

THIS paper describes the progress of research in the corrosion of iron return mains of steam heating systems at the Michigan College of Mining and Technology. The test equipment represents, on a small scale, the ordinary house heating system with boiler, radiator and piping, all of commercial iron and steel.

Arrangement of Test Apparatus

The test equipment has been described in detail in previous papers[1] before this Society. The arrangements of the parts of the system and points at which changes

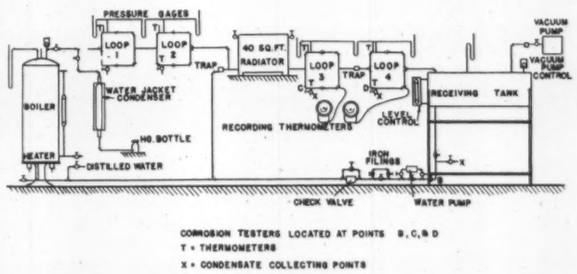


Fig. 1—Line diagram of corrosion test apparatus

in testing methods have been introduced are shown in Fig. 1.

Condensate is removed from Loops 3 and 4 and from the return line, and a sample of steam is obtained. These four samples are analyzed daily for oxygen, carbon dioxide, and pH. The standard Winkler method of oxygen analysis is still being used. The pH is determined by means of a quinhydrone set. The carbone dioxide is determined in two ways: (1) by direct titration with barium hydroxide to the phenolphthalein end point (Dr. Guernsey); and (2) by the evolution method as described by Schroeder and Fellows.[2] The latter method involves acidifying a 200-cc sample of the condensate, heating it and driving off all the carbon dioxide, absorbing the carbon dioxide in barium hydroxide and back titrating the excess hydroxide with hydrochloric acid.

The condensate from Loops 3 and 4 is removed by the mercury bottle method as described in the previous paper. The return line sample is allowed to flow into a flask which is immediately stoppered. Because there is no gas phase above the sample and because the water runs so rapidly, there is not time for the sample to be affected by the atmosphere.

The method of obtaining the steam sample has been changed. The apparatus consists of a water-cooled condenser mounted on a swivel on the side of the boiler, Fig. 2. The inner section is filled with mercury by raising a leveling bottle connected to the bottom of the condenser. When the condenser is full of mercury and all air is excluded, the clamp is adjusted and the condenser is attached to a petcock on the top of the boiler by means of a rubber tube. The petcock and clamp are opened and the mercury leveling bottle is lowered, thus pulling the

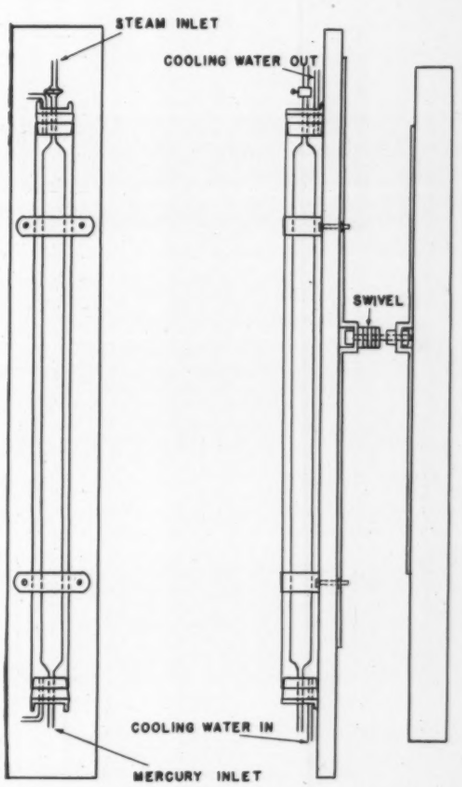


Fig. 2—Glass steam sampling apparatus

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**Research Chemist, Michigan College of Mining and Technology.
[1]Corrosion Studies in Steam Heating Systems, by R. R. Seeber, F. A. Rohrman, and G. E. Smedberg, A. S. H. V. E. TRANSACTIONS, Vol. 40, 1934, and Vol. 42, 1936.
[2]*A. S. M. E. Transactions*, Vol. 54, R. P. 54-13, p. 213. 1932.
For presentation at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1937.

steam in from the boiler. The steam is condensed and, when enough has been obtained for an analysis, the petcock and clamp are closed. The mercury bottle is then raised and the condenser is rocked on the swivel to dissolve any gas which may be present. Finally, the apparatus is disconnected from the petcock and the condensate forced out into a flask by raising the mercury bottle and opening the clamp. While this method is superior to others tried, it is still not ideal. To be truly representative of the steam, a sample must condense all the steam and absorb all the gases associated with the water vapor.

A sample of the condensate taken from below the receiver (point *B*, Fig. 1) would seem to most nearly represent the average steam. The water and the gases in solution flow from this point about 6 ft and then enter the boiler, where they form the steam used.

The steam sample taken from the top of the boiler by the glass tube condenser and mercury bottle is analyzed each day for every test.

The first objective has been to relate the corrosion rate (measured in inches penetration per year times 1,000) to the oxygen and carbon dioxide concentration in the condensate at the three test points, and if possible to relate these concentrations of oxygen and carbon dioxide in the condensate to the amounts of the same gases present in the steam.

Table 1—Tabulation of Corrosion Test Data

Test No. 61

Return Line				Loop No. 4				Loop No. 3			
Time Hours	O_2 cc/l	CO_2 ppm	Temp. F	Time Hours	O_2 cc/l	CO_2 ppm	Temp. F	Time Hours	O_2 cc/l	CO_2 ppm	Temp. F
0.3	4.97	11.4	90	1.6	0.89	8.6	153	2.6	0.50	1.4	177
25.5	3.61	12.2	93	28.8	0.89	11.2	153	30.3	0.40	1.0	177
52.8	3.58	12.9	91	54.3	0.99	8.6	153	55.3	0.27	1.4	177
76.1[a]	3.21	12.2	88	77.3[a]	0.79	10.3	153	78.8[a]	0.33	1.4	177
97.3	3.38	12.2	84	97.9	0.93	10.0	153	98.3	0.36	1.4	177
125.3	2.95	12.2	91	126.3	0.70	10.0	153	127.3	0.23	1.4	177
143.8	2.38	11.7	88	145.3	0.33	10.0	153	146.3	0.36	1.4	177
168.1[b]	2.22	12.2	88	172.6[b]	0.53	11.4	153	173.6[b]	0.27	2.1	177
191.8	1.82	11.4	88	193.3	0.56	10.0	153	194.1	0.23	1.4	177
215.7	1.39	11.4	90	217.3	0.46	10.0	153	218.1	0.33	1.4	177
244.3[c]	0.93	10.7	95	246.1[c]	0.27	10.0	153	246.8[c]	0.43	1.0	177

[a] 9 liters H_2O added—6.06 cubic centimeters per liter O_2.
[b] 9 liters H_2O added—1.95 cubic centimeters per liter O_2.
[c] 9 liters H_2O added—1.46 cubic centimeters per liter O_2.

Test No. 62

2.0	1.13	10.0	90	3.3	0.99	7.9	152	4.8	0.30	1.4	176
21.0	1.36	9.3	91	23.3	0.76	7.9	152	23.3	0.36	1.4	176
46.5	1.26	7.2	84	46.8	0.50	7.9	152	.....	.....	.....	.....
69.8[d]	1.36	8.6	86	70.5[d]	0.50	7.9	152	.....	.....	.....	.....
97.5	0.93	8.6	91	99.0	0.43	7.9	152	100.3	0.36	1.4	176
121.5	0.73	8.6	91	124.5	0.33	7.9	152	124.8	0.46	1.4	176
145.9	0.50	9.3	90	145.8	0.30	7.1	152	148.1	0.36	1.4	176
170.0	0.60	8.6	91	171.5	0.53	8.3	152	172.5	0.36	1.4	176
189.0[e]	0.33	8.9	93	193.5[e]	0.43	8.6	152	194.8[e]	0.56	1.4	176
214.5	0.36	8.6	84	214.3	0.23	7.1	152	.....	.....	.....	.....
243.0[f]	0.33	8.6	93	243.3[f]	0.40	7.1	152	243.6[f]	0.60	1.4	176
261.2	0.20	8.6	91	262.3	0.27	7.1	152	263.3	0.23	1.4	176
286.8	0.17	7.9	90	287.5	0.20	7.1	152	290.0	0.36	2.1	176

[d] 9 liters H_2O added—3.63 cubic centimeters per liter O_2.
[e] 9 liters H_2O added—1.00 cubic centimeters per liter O_2.
[f] 9 liters H_2O added—0.86 cubic centimeters per liter O_2.

Test No. 64

4.0	3.75	2.9	77	5.5	5.05	6.5	91	.....	.....	.....	.....
23.8	5.28	7.9	73	25.0	4.40	8.6	87	5.0	4.86	7.2	105
52.0	5.41	7.2	79	53.3	3.91	8.6	90	30.0	3.89	7.2	107
76.4	5.51	7.2	77	77.5	2.58	7.2	100	54.5	2.51	7.2	120
95.4	4.96	7.2	77	96.6	2.25	6.5	106	73.5	1.79	5.8	133
124.7	4.11	7.2	82	124.7	1.63	6.5	120	101.6	1.04	5.0	147
143.5	3.94	8.6	84	145.0	1.79	6.5	127	126.2	1.34	4.3	156
172.3	3.10	8.3	88	173.8	1.27	7.2	127	150.5	0.72	5.0	157
191.3	2.54	8.6	97	192.3	0.59	7.2	145	169.3	0.49	2.2	170
220.0	2.64	6.5	93	221.3	1.04	7.2	127	198.0	0.36	3.6	152
239.5	2.77	5.8	88	241.5	0.95	7.2	127	218.3	0.68	3.6	156
262.8	2.41	7.2	86	264.8	0.59	5.8	135	239.8	0.49	3.6	170

Test No. 51

20.7	2.44	8.8	.....	24.0	2.14	9.5	91	25.0	1.97	9.9	105
45.7	2.49	8.8	.....	48.0	1.93	8.8	98	49.2	1.55	8.8	111
69.5	2.44	8.8	.....	71.0	1.85	10.2	103	77.1	1.51	7.3	119
93.1	2.40	8.8	.....	94.8	1.81	10.2	100	96.0	1.22	7.3	120
142.3[g]	2.11	8.8	.....	147.0[g]	1.72	8.8	95	148.0[g]	1.47	10.2	110
165.9	2.02	8.8	.....	170.8	1.55	10.2	100	172.9	1.43	7.3	117
190.0	2.02	8.8	.....	192.5	1.26	9.1	104	194.0	1.05	8.5	125
212.9	1.85	8.8	.....	214.8	1.09	8.5	105	219.0	1.22	7.3	130

[g] 12 liters H_2O added—6.02 cubic centimeters per liter O_2.

The difficulties to be overcome are due principally to the great number of causes which may affect the corrosion rate. For a system of this sort using distilled water, the following list is believed to include all those causes which produce any appreciable effect on the corrosion rate: oxygen, carbon dioxide, temperature, rate of flow, character of the specimen, pH, diffusion rate and diffusion constants, pressure, and the gaseous phase. By keeping the pressure very nearly constant and the surrounding atmosphere at a constant temperature, it has been possible to hold within narrow limits the variations of the temperature and rate of flow at the various points of the test.

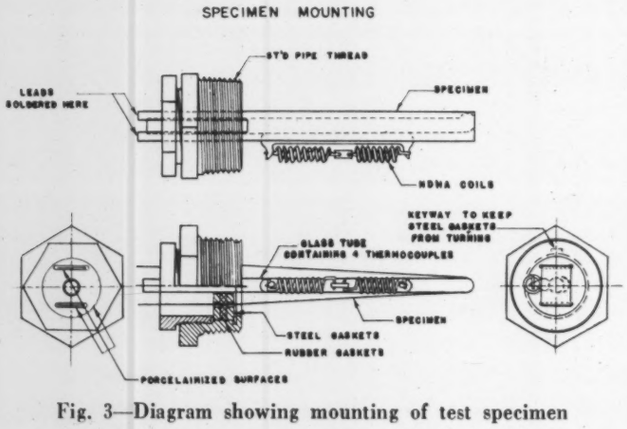

Fig. 3—Diagram showing mounting of test specimen

The pH value under the conditions of tests during the past year has varied but little, averaging about 5.5. Since the corrosive action on the test specimen takes place under a film of corrosion product (presumably FeOH) the actual pH at this point is more nearly related to the pH of this corrosion product than to the pH of the surrounding solution until the latter becomes strongly acid (below pH 4) or strongly alkaline (above pH 8).

Speller says: "With waters in the neutral range, the

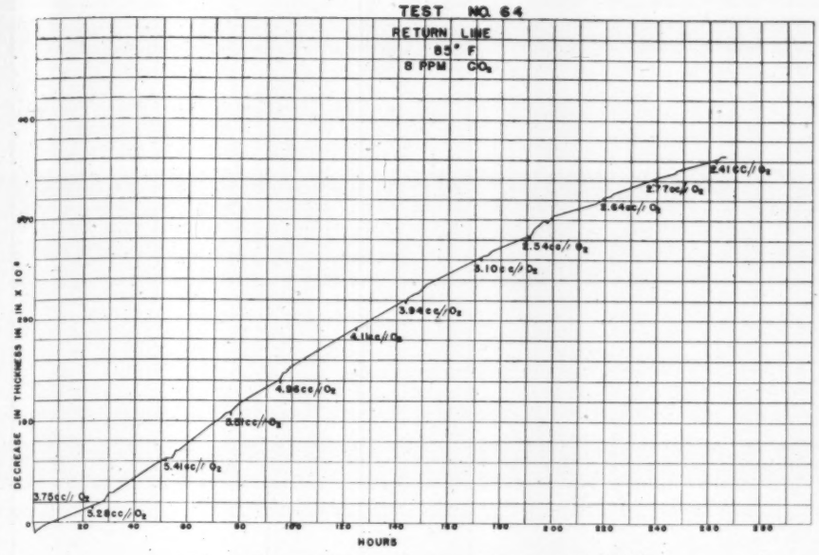

Fig. 4—Curve showing decrease in test strip thickness and oxygen analysis of samples in return line

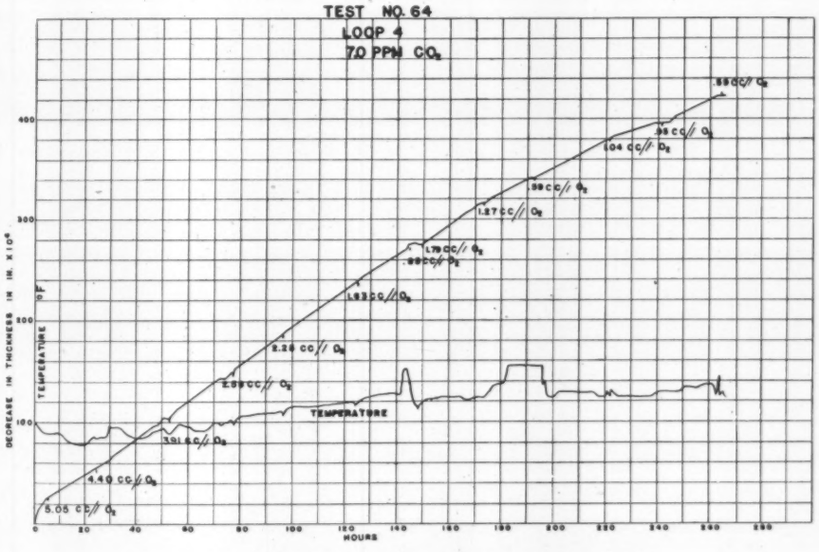

Fig. 5—Curve showing decrease in test strip thickness and oxygen analysis of samples in Loop No. 4

surface film tends to remain at a pH of 9.5 regardless of the pH of the water as there is not enough acid diffusing in from the main solution to neutralize the surface alkalinity. If, however, the main solution is strongly alkaline, the film itself becomes more alkaline, the solubility of ferrous hydroxide is decreased, and the hydroxide is precipitated more rapidly with the formation of a more protective rust coating. This explains the action of alkalies in reducing corrosion. Conversely, if the main solution is sufficiently acid, the acid diffuses in toward the metal rapidly enough to dissolve the protective ferrous hydroxide rust coating and to destroy the alkalinity of the film. Rapid corrosion can then occur, both because the protective coating is not present and because hydrogen gas can be generated from the more acid solution against the metal."[3] Whitman, Russell, and Alterie agree with this.[4]

[3]Speller, Corrosion, Causes and Prevention, 2nd Ed., p. 162.

[4]Whitman, Russell and Alterie, *Ind. & Eng. Chem.*, Vol. 16, p. 665, 1924.

The introduction of the electrical resistance method of measuring corrosion rates as described in the previous paper[5] presented before the Society, and the *American Society for Testing Materials*, January, 1937, has made possible a much greater number of corrosion rate observations without stopping the tests to remove specimens. While standard *National District Heating Association* test coils have been exposed in each test in the same location as the resistance test strips, for this discussion only electrical resistance rates are used. The mounting of specimen is shown in Fig. 3.

The testing system is a closed system: the condensate returns to the boiler, and the only change in the liquid contents is caused by the removal of samples for chemical analysis and the occasional addition of water to the boiler. The system is practically tight. Before starting a test a high vacuum is produced, which changes only a fraction of an inch in 24 hours. All tests from which data, given in Table 1, for this paper were taken were at atmospheric pressure so that the gaseous phase was changed only by the progress of the test.

The test strips are made from cold-rolled Bessemer strip steel furnished by the U. S. Steel Co. through their Cuyahoga Works at Cleveland; it has been analyzed by them as:

Carbon	0.10
Manganese	0.30
Sulfur	0.037
Phosphorus	0.077

or almost exactly the analysis of the *National District Heating Association* test coils. Although these test pieces are prepared by exact routine with an annealing by electricity in a vacuum for each, it is not to be expected that the resistance to corrosion of all pieces is precisely the same. Many tests under similar conditions should indicate the mean. The unit resistance of these specimens varies but little.

The decrease in thickness of the test strip with time is given in Fig. 4 and also the O_2 analysis of samples taken at the various times indicated by the position of the analysis on the chart. This plate gives results from one test on condensate from the receiver and the analysis represents most nearly the chemical constituents of the steam at the times given.

Fig. 5 charts similar information obtained from Loop No. 4, when a lower temperature than usual was desired and it became difficult to maintain this temperature at a constant figure. The temperature variations were there-

[5]Loc. Cit. See Note 1.

fore plotted to scale on the chart. Similar charts were prepared for each test, and from the corrosion rates obtained from these and the corresponding chemical analyses, Figs. 6 and 7 were plotted.

Most of this work was done with CO_2 concentrations of about 10 parts per million. Similar work must be done

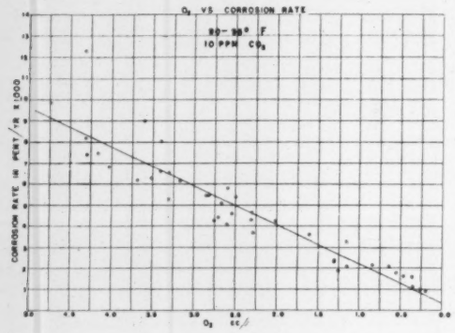

Fig. 6 — Corrosion rate plotted with reference to the oxygen concentration over a temperature range from 80 to 95 F

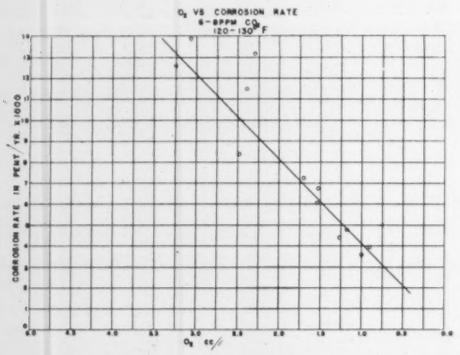

Fig. 7—Corrosion rate plotted with reference to the oxygen concentration over a temperature range from 120 to 130 F

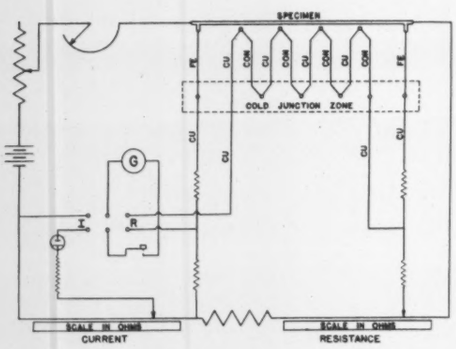

Fig. 8—Complete electrical circuit diagram for mounting of a resistance specimen

with other concentrations of CO_2 to enable deductions to be made as to the probable effect of changes in this constituent.

Assume a test starting with 1.0 cc per liter of O_2 in the condensate in the receiver (test point 3). As the test proceeds, this oxygen is reduced by combination with the iron of the system and approaches a value of zero cc per liter. If samples for analysis are taken at intervals and a curve is plotted of reduction in thickness of test specimen against time (Fig. 4), a line connecting two consecutive points of sampling (shown at intersection of sampling time with curve) will show the rate of decrease of thickness and consequently the inches penetration per year, while the average of the two analyses will indicate the oxygen concentration related to this rate.

From plotted results obtained in this manner for various O_2 concentrations a curve showing change of corrosion rate with change of O_2 concentration may be approximated. In tests thus far this year CO_2 was also present. Where the temperatures are not too great, as in Loop 4 and in the return main, many tests have shown consistently that, while the oxygen decreases with lapse of time, the CO_2 concentration remains substantially constant. As the CO_2 remains constant the only variable is the oxygen, and curves can be drawn for the corrosion rate of O_2 in the presence of a given amount of CO_2 and at constant temperature and at a definite rate of flow of condensate past the test specimen.

The test results plotted (Figs. 6 and 7) indicate that the corrosion rate increases directly with the O_2 concentration, CO_2 being constant. With the same concentrations of gases and higher temperatures, the corrosion rates are much higher.

Although the plotting of test results shows wider individual variation from a mean than is desirable, the inferences drawn appear to be justified. These variations in results seem to have been caused by actual variations in some of the many variables which it was attempted to maintain constant throughout the test periods. While the methods of analysis are quite accurate, it is unlikely that the samples taken are at all times truly representative of the liquid in contact with the test specimen.

One interesting step will be to reduce the O_2 concentration to zero and then note the effect of the CO_2 acting alone.

Although definite information as to the effect of temperature change on the rate of corrosion with constant O_2 and CO_2 concentration in the condensate can be expected from a continuation of this research, this information might be more rapidly obtained from a laboratory testing equipment such as used by Finnegan, Corey and Jacobus.[6]

The effect of variations of rate of flow or velocity of the stream of condensate passing the test specimen should be carefully investigated. This research might well be combined with the investigation of the effect of temperature changes.

The electrical resistance method of corrosion testing has been improved during the past year. Variations in observed resistances were found to be due to variations in contact resistance between the leads. A new method of making these contacts reduced variations materially. A new mounting of the resistance specimen as shown in Fig. 8 reduced troubles from possible contact of resistance strip with *National District Heating Association* testers or the plug itself.

Acknowledgments

Acknowledgment is due to G. H. Damon of the Chemistry Department, Michigan College of Mining and Technology, for much advice and assistance, and to G. H. Lyle of the Electrical Engineering Department, Michigan College of Mining and Technology, for assistance in improving the equipment for electrical resistance measurements.

[6] *Ind. Eng. Chem.* 27, p. 774, 1935.

Study of Summer Cooling in the Research Residence Using Water at Temperatures of 52 F and 46 F

(Part 2)

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Urbana, Ill.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the *National Warm Air Heating and Air Conditioning Association* and the University of Illinois

Acknowledgments

The results presented in this paper were obtained in connection with the summer cooling investigation (1936) in the Research Residence‡ at the University of Illinois, conducted by the Engineering Experiment Station of which M. L. Enger, Dean of the College of Engineering, is the director, in the Department of Mechanical Engineering of which O. A. Leutwiler, Professor of Mechanical Engineering Design, is the head. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is also due to M. K. Fahnestock, Research Assistant Professor, and R. B. Engdahl, Research Assistant, for services rendered in connection with the investigation, and to the various manufacturers who cooperated by furnishing instruments and equipment.

Actual Cooling Loads

During the periods of intermittent operation the actual cooling load curve showed high peaks at the start of each operating period. These peaks correlated with periods of high indoor relative humidity, indicating that periods of high wet-bulb temperature for the air entering the coil were accompanied by corresponding periods of comparatively high heat transmission for the coil. Similar conditions were observed in the tests made in 1935 with 58 F water supplied to the cooling coil. It was shown that the heat absorbed by the cooling coil was practically a linear function of the enthalpy of the entering air. Furthermore, periods of high indoor relative humidity also represented periods of high wet-bulb temperature with corresponding high enthalpy for the air entering the coil. Since high heat transmission accompanied high enthalpy for the air entering the coil, it serves to explain the fact that the peaks in the load curve in Fig. 4 occurred at times corresponding to high indoor relative humidity.

In this connection it is of interest to compare the typical results which were obtained from the tests made in 1935 and 1936. In 1935 the indoor dry-bulb temperature was maintained at approximately 80 F for a day in which the maximum outdoor temperature was 96 F, giving a temperature difference of 16 F. The heat absorbed by the 58 F water passing through the cooling coil was 23,700 Btu per hour. In 1936 the indoor dry-bulb temperature was maintained at approximately 80 F on a day in which the maximum outdoor temperature was 104 F, giving a temperature difference of 24 F, and the heat absorbed by the 52 F water passing through the cooling coil was 24,150 Btu per hour. That is, for an increase in temperature difference between the maximum outdoor temperature and the average indoor temperature of 8 F or 50 per cent, the increase in the rate of heat absorption by the cooling coil was only 2 per cent.

This may be accounted for by the fact that in 1935 the relative humidity of the air entering the cooling coil was 62.8 per cent, whereas in 1936 the value was 48 per cent. If the air entering the cooling coil had contained more moisture in 1936, with a correspondingly higher wet-bulb temperature and enthalpy, then since the heat transmission is practically a linear function of the enthalpy of the entering air, a much greater rate of heat transmission would have been obtained. This would probably have been sufficient to compensate for the resulting increase in the latent heat load. The fact that the absorption of sensible heat was only 19 per cent greater in 1936 than it was in 1935 while the increase in sensible heat load was 50 per cent makes it probable that the plant was overloaded in 1936. Further evidence on this point is given by the fact that, as shown in Fig. 4, the indoor temperature was increasing at 2:00 p.m., and continued to do so up until 6:00 p.m. The total rise, however, was only approximately 1.5 F and had no appreciable influence on the comfort conditions maintained. Thus the heat lag of the structure tended to offset the influence of the higher outdoor temperature occurring in 1936, by preventing an abnormal rise in indoor temperature even though the plant was not able to completely carry the load.

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‡The Research Residence in Urbana, Ill., was built, furnished, and completely equipped specifically for research work in warm air heating by the *National Warm Air Heating and Air Conditioning Association* in December, 1924.

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For presentation at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1937.

Calculated Cooling Load

The total cooling load imposed on a cooling plant consists of the sum of all heat gains by the structure resulting from indoor-outdoor temperature differences and sun effects on walls and windows, and of all other heat quantities introduced by air inleakage and heat given up by the occupants and by mechanical or electrical processes. These heat quantities may be itemized as follows:

Item No.	Heat Source
1	Heat transmission through walls not exposed to sun
2	Heat transmission through walls exposed to sun
3	Heat transmission through windows and doors not exposed to sun
4	Heat transmission through windows exposed to sun
5	Heat transmission through ceilings
6	Heat transmission through floors
7	Heat brought in by infiltration or ventilation air
8	Heat given off by occupants
9	Heat given off by electric motors
10	Heat given off by electric lights

Complete details of the method of calculation of the cooling load have been outlined in a previous publication[5] and will not be repeated in this paper. The values of coefficients of heat transmission and enthalpies of mixtures of water vapor and air as given in The American Society of Heating and Ventilating Engineers Guide, 1936, have been used.

The most common method of calculating the cooling load, which changes from hour to hour depending on variations in outdoor temperature and sun effect, is based on the assumption that the changes in load occur coincident with the changes in outdoor temperature and sun effect.

If, for example, the temperature difference between the outdoors and indoors is 20 F at a given time, the heat transmission through the wall at that time is calculated on the basis that conditions of thermal equilibrium have been maintained with this 20 F difference and the existing sun effect. However, since the walls offer some resistance to heat flow, changes in outdoor temperature and sun effect are not reflected instantaneously in changes in heat flow, and an appreciable interval exists between the time that a given outdoor temperature and sun effect is imposed and the time at which the heat reaches the indoors. This becomes evident from the fact that the temperature of the inside surface of the wall reaches a maximum considerably later than that attained by the outdoor temperature.

A second method for calculating the cooling load takes into account this time lag in the transmission of heat through the walls and ceiling. In this case the calculated value of the heat transmission through the wall at any given hour is based, not on the temperature difference between the outdoors and indoors and the sun effect occurring at the given time, but on the temperature difference and sun effect that existed at some previous time. Obviously, the lag that exists between the time that a temperature differential is imposed on a wall and the time that the inside surface temperature is affected is dependent on the type of construction, the mass, and the conductivity of the wall. A massive wall that is well insulated, for example, will show a greater time lag than a lighter wall that is not as well insulated.

[5]University of Illinois, Engineering Experiment Station, Bulletin No. 290, 1937, A. P. Kratz, M. K. Fahnestock, and S. Konzo.

In order to determine the difference in the results obtained with the two methods of calculating the cooling load, the hourly values for the cooling load were first determined by assuming no time lag. These hourly values of the heat transmission were calculated for July 7, 1936. The values of the ten individual items of heat gain are represented in Fig. 5a and the total calculated cooling load obtained by summing the values of the individual items at any given time is shown in Fig. 4. It may be noted from Fig. 5a that the maximum value for the heat transmission through the walls which were not exposed to the sun (Curve No. 1) was obtained at approximately 4:30 p.m. Similarly the maximum values for the items represented by curves numbered 2 and 5, respectively, were obtained at 2 p.m. and 3:30 p.m.

For the second method of calculating the cooling load the amount of the time lag to be used was determined from experimental data obtained by means of thermocouples placed on the inside surfaces of the walls and ceiling. These studies indicated that for the conditions maintained in the Research Residence an interval of approximately 2 hours was required for heat to flow through the sidewalls and an interval of approximately 3 hours for heat to flow through the second story ceiling. Hence the heat gains represented by curves Nos. 1 and 2 in Fig. 5a were not changed in magnitude, but were replotted against a new time basis in Fig. 5b so that the maximum values were obtained 2 hours later. Similarly the curve representing the heat transmission through the ceiling, curve No. 5a, was replotted in Fig. 5b so that the maximum value was obtained 3 hours later than that shown in Fig. 5a. It may be noted from Fig. 5b that the maximum values for the items represented by curves Nos. 1, 2, and 5 were therefore obtained at 6:30 p.m., 4 p.m., and 6:30 p.m., respectively. The seven remaining items of heat gain were not influenced by heat lag and hence the curves representing these items in Fig. 5b were not changed as compared with the corresponding curves shown in Fig. 5a. The total calculated cooling load was again obtained by summing the individual items in Fig. 5b at any given time and the resulting curve is shown in Fig. 4.

It may be observed from the two curves shown in Fig. 4 for the calculated cooling load on the Residence that the maximum values obtained by the two methods of calculation were practically the same, although the times at which the maximum values were obtained were approximately 1 hour apart. Although the maximum values were obtained at approximately 3 p.m. when no time lag was assumed and at approximately 4 p.m. when a time lag was assumed, the actual temperature of the indoor dry bulb did not attain a maximum value until approximately 6 p.m. This would tend to indicate that in order to obtain a closer correspondence between the calculated cooling load and the actual cooling load, an even greater time lag than that which was used would have to be assumed. However, the use of a greater time lag in the case of the Research Residence could not be justified from the observations of the inside surface temperatures of the walls and ceilings.

In the calculation of the design load the maximum value of the load is of more consequence than is the time at which the maximum occurs. The fact that practically the same maximum value was obtained by both methods

of calculation seems to indicate that in the case of frame structures similar to the Research Residence the more complicated method of calculation based on the assumption of time lag is hardly justified, although it may be justified for more massive structures.

The curve based on the assumption of the existence of a time lag showed, in general, a more gradual rate of increase of the cooling load until the maximum value was reached and also a less rapid decrease of the cooling load after the maximum value was attained than was shown by the curve based on the assumption of the existence of no time lag. It should be noted that the actual heat absorption during this entire period remained constant in spite of the fact that the indoor dry-bulb temperature slightly exceeded the predetermined value of 80 F. Hence, if the capacity of the cooling plant had not been limited and if the actual rate of heat absorption by the coil could have been increased by increasing the rate of water flow so that the indoor temperature had been maintained at a constant value of 80 F during the period from 2 p.m. to 10 p.m., it seems likely that the curve representing the actual heat absorption would have attained a closer correspondence to the calculated curve based on the assumption of time lag than to the curve based on the assumption of no time lag. Under these conditions, it is possible that the method of calculation based on the assumption of a time lag may have some merit and consideration of it should not be entirely neglected in design procedure,[6] particularly in the case of massive structures.

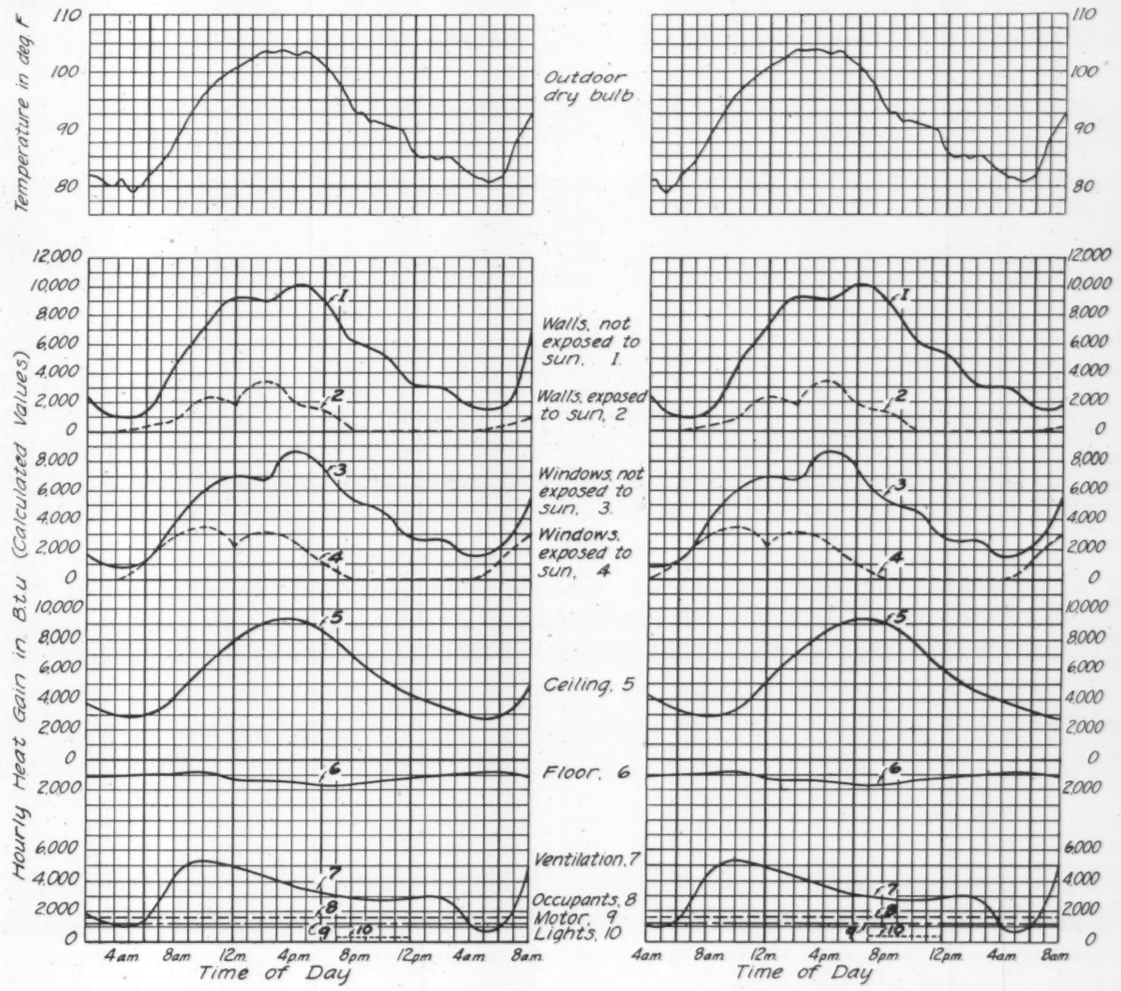

Fig. 5a—Calculated cooling load, no time lag assumed. Calculated components of cooling load on residence, July 7, 1936, Test No. 12, Series 1-36, maximum outdoor temperature 104 F

Fig. 5b—Calculated cooling load, time lag assumed. Calculated components of cooling load on residence, July 7, 1936, Test No. 12, Series 1-36, maximum outdoor temperature 104 F

Possible Reductions in Cooling Load

The curves shown in Fig. 5b are of additional interest in that they present in a graphical manner the relative magnitudes of the ten component items that comprised the total calculated cooling load on the Residence on a typical day, in which the outdoor temperature attained a maximum value of 104 F. The maximum values attained by each component item, irrespective of the times at which they occurred have been listed in Table 3.

[6] A Rational Heat Gain Method for the Determination of Air Conditioning Cooling Loads, F. H. Faust, L. Levine, and F. O. Urban, A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, Vol. 7, No. 8, August, 1935, pp. 391-401.

Table 3—Maximum Calculated Values for the Ten Component Items of the Cooling Load

Based on Test No. 12, July 7, 1936. Research Residence

Item No.	Source of Heat Gain	Maximum Value Btu per Hour	Per Cent of Total of Maximum Values
1	Walls, not exposed to sun	10,100	24.2
2	Walls, exposed to sun	3,400	8.2
3	Windows, not exposed to sun	8,700	20.9
4	Windows, exposed to sun	3,500	8.4
5	Ceiling	9,300	22.3
6	Floor	—1,700	—4.1
7	Ventilation	5,300	12.7
8	Occupants	1,600	3.8
9	Motor	1,200	2.9
10	Lights	300	0.7
	Total[a] of maximum values	41,700	100.0

[a]The total of the maximum values of the component items is approximately 13 per cent greater than the calculated value of the maximum cooling load that was obtained at 4 p. m.

For this particular case, practically one quarter of the total value may be attributed to the heat transmission through the walls which were not exposed to the sun and approximately one twelfth to the heat transmission through walls which were exposed to the sun. That is, approximately one third of the total may be attributed to the heat transmission through the walls alone. It may be observed that by the application of effective insulation to the uninsulated side walls, for which the coefficient of heat transmission was 0.25 Btu per square foot per hour per degree difference in temperature, substantial reductions in the heat gain would be possible. This reduction in the heat gain through the walls themselves would be proportional to the reduction in the coefficient of heat transmission. The percentage reduction in the total heat gain would be one third of that through the walls alone. In addition, the time lag for the passage of heat through the walls would also be increased so that the cooling load would be distributed more uniformly with respect to the time of day. In a similar manner substantial reductions would be possible in the heat transmission through the windows which were not exposed to the sun, provided double-glass windows could be used in place of the single-glass windows. Since all of the windows exposed to the sun were equipped with awnings, the value listed as item 4 was considerably smaller than that which would have been obtained had no awnings been used.

It may be observed from Table 3 that the maximum heat gain through the walls exposed to the sun was 3,400 Btu per hour, or only 8.2 per cent of the sum of all maximum heat gains. The indoor-outdoor temperature difference occurring at this time was 23.4 F, and the total area exposed to the sun was 463 sq ft. If the sun effect at this time had been neglected, and the heat transfer calculated as merely that resulting from the temperature difference, a value of 2,720 Btu per hour would have been obtained. Comparing this with the 3,400 Btu per hour obtained by computations involving the sun effect, it is evident that the difference is only 680 Btu per hour, or 1.6 per cent of the sum of all maximum heat gains. Since this is within the limits of accuracy of any of the methods used for computing heat gains, it would seem that, for the purpose of obtaining the maximum load to be used for design, the more involved method taking into account the sun effect on the walls can hardly be justified in the case of a structure similar to the Research Residence. This is not true of windows exposed to the sun, however, because if the windows had not been shaded by awnings, the resulting heat gain would have been approximately 12,100 Btu per hour instead of the 3,500 shown in Table 3, and would have constituted a large portion of the total heat gain. Hence the sun effect on windows, and particularly on those which are unshaded, cannot be neglected.

The amount of heat transmitted through the ceiling (Item 5) is exceedingly large, due not only to the relatively high coefficient of heat transmission, but also to the very large temperature difference that exists between the attic space and the indoor air. The effective use of insulation should serve to materially reduce the heat gain from this source also. The heat brought in by ventilation and infiltration (Item 7) is dependent not only on the amount of air introduced into the house but also on the climatic conditions that exist. In this connection, it may be noted that when the moisture content of the outdoor air is high as was the case in 1935,[7] the numerical value of Item 7 may be much greater than that listed in Table 3. Restricting the amount of air required for ventilation tends to accentuate any odors, and further reduction in the number of air changes may not be advisable in the average residence. The values listed as Items 8, 9, and 10 are usually insignificant in the case of residence installations, although they may be of considerable importance in commercial or industrial installations. An analysis of the components of the calculated cooling load therefore indicates that the size of the cooling unit required and the cost of operation of the unit may be materially reduced by substantial improvements in the construction of the house. The use of awnings on all windows exposed to the sun, of double-glass in windows, and of effective insulation both in the sidewalls and in second story ceilings or attic floors should be encouraged if summer cooling of residences is to become a practical matter.

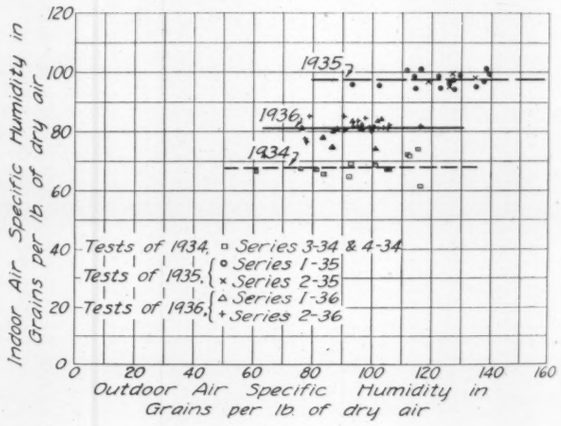

Fig. 6—Specific humidity of indoor air resulting from operation with two types of cooling systems

Humidity Conditions Maintained.

The curves in Fig. 6 show that during periods of continuous operation, both in the case of the water coil used in the studies made in 1935 and 1936 and in the case of the mechanical refrigerat-

[7]The load due to ventilation at 2 p.m. on August 2, 1935, was 8715 Btu per hour.

ing unit used in the study of 1934, the indoor specific humidity maintained was dependent only on the temperature of the medium used and the character of the cooling coil and was independent of the amount of water vapor in the outdoor air. Since in both cases a constant indoor dry-bulb temperature of 80 F was maintained, the specific humidity is also representative of the indoor relative humidity. That is, the water coil used in 1935 with a mean water temperature of approximately 62 F maintained a constant indoor relative humidity of 63 per cent during periods of continuous operation, while the water coil used in 1936 with a mean water temperature of 57.6 F and 55.5 F for Series 1-36 and Series 2-36 respectively, maintained a constant relative humidity of approximately 52 per cent for both cases. The evaporator coil used in 1934 with a mean temperature of approximately 45 F for the refrigerant maintained a constant relative humidity of 45 per cent.

Possibly by coincidence the numerical value of the relative humidity that was maintained in the house was nearly the same as the numerical value of the mean coil temperature. Ordinarily for small cooling installations, in which the selection of the size of the cooling coil is governed by considerations based only on the maintenance of some given value of the indoor dry-bulb temperature, the indoor relative humidity accompanying this temperature is more or less indeterminate. Unless special efforts are made either to dehumidify the air by the use of additional equipment, or to reheat the air either directly or otherwise, the indoor relative humidity that will be obtained cannot be maintained at any arbitrarily selected value. Over a limited range of mean coil temperatures and under test conditions similar to those maintained in the Research Residence, the suggested approximation may be of some value in determining the probable magnitude of the indoor relative humidity in similar residence installations. On the assumption that this relationship is sufficiently general until proved otherwise, it might serve as an index for the selection of desirable coil temperatures.

During periods of operation the ventilating air from outdoors was mixed with the recirculated air from the house just before entering the cooling coil. The moisture content of the outdoor air therefore directly influenced that of the mixed air entering the coil. In the tests reported in 1935 it was indicated that the enthalpy of the entering air increased as the specific humidity increased, resulting in an increase in the heat transmission for the coil. At the same time the specific humidity of the air leaving the coil increased, but not to as great an extent as that of the inlet air. This indicated that a part, if not all, of the increase in heat transmission was utilized in increasing the amount of condensation as the specific humidity of the outdoor air and hence that of the air entering the coil, was increased. Thus the increasing heat transmission of the coil tended to compensate for increases in the outdoor specific humidity, and, as a result, also tended to maintain a constant indoor relative humidity during periods of continuous operation, as shown in Fig. 6.

The maintenance of a practically constant indoor relative humidity accompanying rather wide variation in the outdoor humidity conditions would imply that the proportion of the total cooling load that could be attributed to the process of dehumidification would increase as the outdoor humidity increased. The curves presented in Fig. 7 indicate that such was the case. It may be observed from the results obtained in the 1936 tests that the ratio of the moisture load to the total load increased from approximately 0.05 to 0.22 when the outdoor specific humidity increased from 80 grains to 100 grains per pound of air. A similar increase may be observed for the tests made in 1935, although both the lowest value and the total range of the outdoor specific humidity were definitely greater in that year, indicating a much wetter season than the one for 1936. From extrapolation of the curve for 1936 in Fig. 7 it is evident that had the 1936 season been characterized by as much moisture in the outdoor air as was the case for the 1935 season, the ratios would have been equal to or greater than those obtained in the 1935 tests. Therefore, it is apparent that the relative magnitude of the load resulting from the moisture condensed was dependent not only on the mean coil temperature used but also on the amount of moisture contained in the outdoor air. Furthermore, since the extrapolated portion of the curve for 1936 would lie above the curve for 1935, it seems probable that the increased heat transfer, resulting from the lower mean coil temperature and the increased enthalpy of the entering air, would have made it possible to have maintained a lower indoor relative humidity in 1936 than was maintained in 1935 even if the 1936 season had been as damp as the one for 1935.

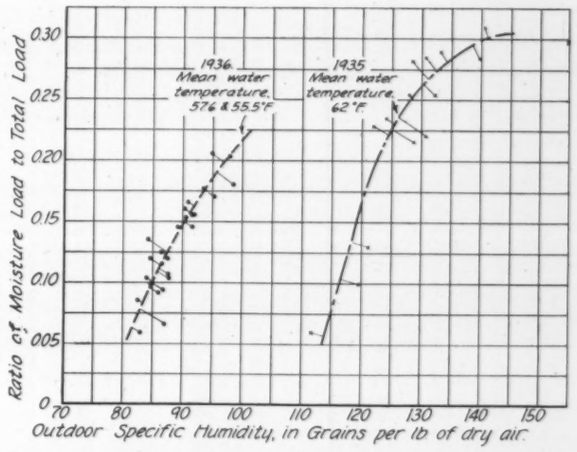

Fig. 7—Ratio of the load due to condensed moisture to the total cooling load

Heat Absorbed per Day

The total cooling load for the day, expressed as the total Btu absorbed from the air in the total amount of time during which water was actually being circulated is shown in Fig. 8 plotted against the degree-hours above 85 F per day. The base temperature of 85 F was selected because it was found that usually no cooling was required unless the outdoor temperature was 85 F or more. Curves for both total heat and sensible heat are shown, and the difference between these curves represents the heat due to the change in the moisture content of the air. Considerable deviations from the curves representing median conditions occurred in the individual points, each of which was obtained from data based on

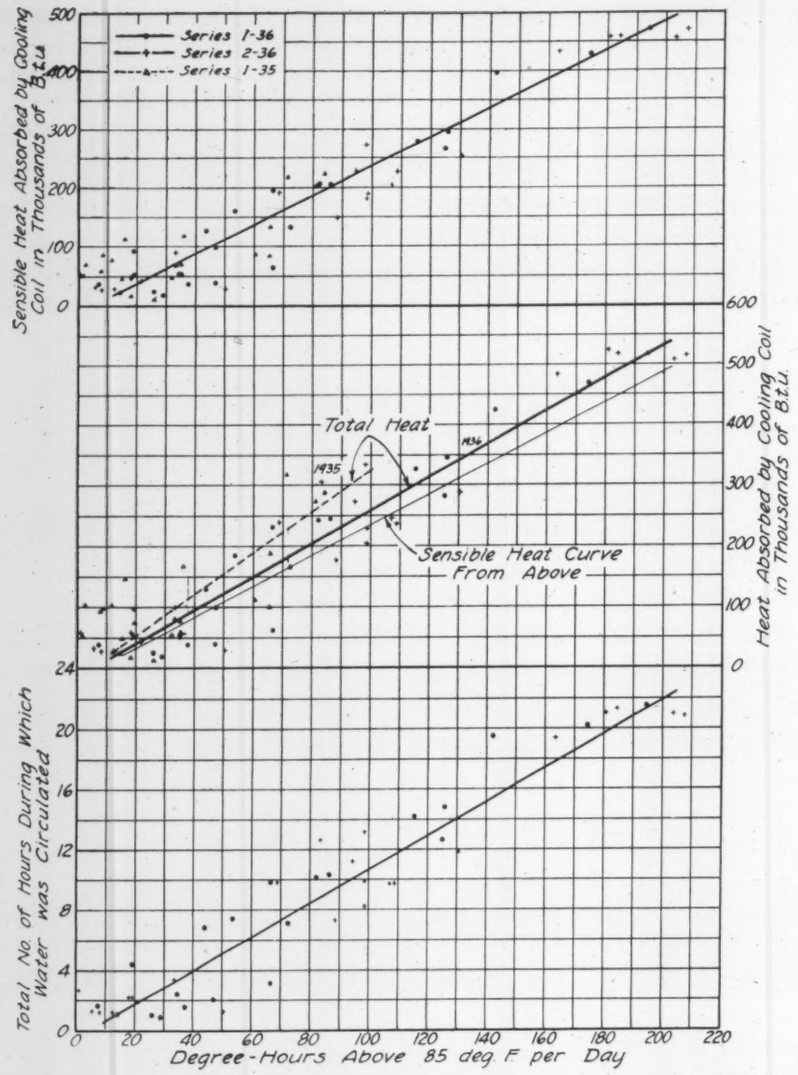
Series 1-36
Series 2-36
Series 1-35
Sensible Heat Absorbed by Cooling Coil in Thousands of B.t.u.
Heat Absorbed by Cooling Coil in Thousands of B.t.u.
Total Heat
1936
1935
Sensible Heat Curve From Above
Total No. of Hours During Which Water was Circulated
Degree-Hours Above 85 deg. F per Day

Fig. 8—Heat absorbed by cooling coils and hours of water circulation per day

a single day's operation. These deviations could be accounted for by variations in the wind and sun effects which influence the load but are not directly dependent upon the degree-hours. In addition, if the night was exceptionally cool, less than the average amount of artificial cooling was required the next day, while if the night was exceptionally warm the reverse was true.

From the upper group of curves in Fig. 8 it may be noted that the sensible heat absorbed during the tests made in 1935 and during the two series of tests made in 1936 could be represented by a single curve. This would indicate that the amount of sensible heat absorbed by the cooling coil for a given outdoor condition is independent of the coil temperatures and is dependent only on the temperature difference maintained between the outdoor air and the indoor air. On the other hand the middle group of curves in Fig. 8 indicates that the total heat absorbed by the cooling coil was greater for the tests made during the 1935 season than for the tests made during 1936. This condition was caused by the fact that a relatively greater amount of heat was required to reduce the moisture content of the air during the former season, which was characterized by a high moisture content in the outdoor air. The curves in the lower portion of Fig. 8 show the total number of hours per day during which water was circulated. For those days on which the outdoor temperature attained a maximum value between 104 F and 108 F the operation of the plant was practically continuous, since the cold water was circulated approximately 22 hours out of the 24. These curves shown in Fig. 8 can be used to estimate the seasonal amount of heat absorption and the total time of plant operation for any season for which the number of degree-hours above 85 F corresponding to each day in the season is known.

A comparison of the sensible heat absorbed by the coil supplied with cold water in 1935 and 1936 with that absorbed by the expansion coil used in connection with the mechanical refrigerating unit employed in 1934 affords a means of determining the magnitude of the heat gain introduced into the house by the presence of the refrigerating machine in the basement. This is true because the heat emitted from the machine would appear as a sensible heat gain alone, and therefore the difference in latent heat loads for the two seasons may be disregarded. In the case of the coil supplied with cold water the amount of sensible heat removed from the air by the water is equal to the net sensible heat gain of the house. However, in the case of the mechanical refrigerating unit located in the basement, the sensible heat gains contributed by the unit, such as the electrical and mechanical losses in the compressor motor, the mechanical losses in the compressor and drive, and the heat losses from the condenser and compressor heads constituted an additional load in excess of the net sensible heat gain of the house. The result was such that on two similar days characterized by the same number of degree-hours the sensible heat absorbed by the direct expansion coil used in connection with the mechanical unit was greater than that absorbed by the water coil. It was observed that for a day having 80-degree-hours, the sensible heat absorbed by the mechanical unit was 250,000 Btu[8] per day and by the water cooling coil was 190,000 Btu per day. The difference of 60,000 Btu per day amounted to 24 per cent of the heat absorbed by the mechanical unit. The ratio of 190,000 to 250,000 may be regarded as an approximate index of the overall house efficiency of the mechanical unit. This ratio of 0.76 agrees remarkably well with the value of 0.767 obtained directly from an analysis of the heat balance on the house, as presented in a previous paper.[9]

[8]A.S.H.V.E. Research Paper, Study of Summer Cooling in the Research Residence for the Summer of 1934, A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick, A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, January, 1935, pp. 29-40.

[9]Loc. cit. (Note 8.)

Table 4—Summary of Results of Tests Using Water for Cooling and Using Night Air Cooling for Entire Season of 1936

Weather Data		
1.	Total hours above 85 F for season of 1936	706.3
2.	Total hours above 90 F for season of 1936	341.6
3.	Total degree-hours above 85 F for season of 1936	4551.0
4.	Total degree-hours above 90 F for season of 1936	1962.8
Night Air Cooling		
5.	Total number of nights with night air cooling by fan	62
6.	Total running time for fan during night air cooling, hours	614.4
7.	Average rate of power input to fan during night air cooling, watts	503
8.	Total power input to fan during night air cooling, kilowatt hour	308.9
Air Recirculation without Water Cooling		
9.	Total number of days with recirculation of house air by fan	63
10.	Total running time for fan during recirculation of house air, hours	621.0
11.	Average rate of power input to fan during recirculation without water cooling, watts	441
12.	Total power input to fan during recirculation of house air without cooling, kilowatt hour	273.5
Air Recirculation with Water Cooling		
13.	Total number of tests with water cooling	51
14.	Total running time for fan during test period, hours	695.0
15.	Average rate of power input to fan during test period, watts	442
16.	Total power input to fan during test period, kilowatt hours	307.5
Combined Totals		
17.	Total running time for fan including night air cooling, hours	1930.4
18.	Total power input to fan including night air cooling, kilowatt hours	887.7
19.	Total number of hours of test period for season	695.0
20.	Total number of hours of water circulation during test period	452.1
21.	Quantity of cooling water	
	Gallons, Series 1	54,991
	Series 2	39,531
	Total	94,522
	Cu ft, Series 1	7329
	Series 2	5271
	Total	12,600

Summary of Seasonal Results

Table 4 gives a summary of the total quantities obtained for the season of 1936. This season was the hottest summer occurring in Urbana, Illinois since the year 1888, when the local Station of the U. S. Weather Bureau was established. As may be noted from Fig. 9 the average of the 122 daily maximum temperatures that were obtained during the months of June, July, August, and September was 88.0 F, which was approximately 1 deg higher than the greatest average of the daily maximums obtained during the previous 49 years. The total of 4551 degree-hours above 85 F, which was almost twice that obtained for 1934, is a better indication of the extreme weather conditions that prevailed during the summer.

A comparison of the weather conditions between 1936 and 1935 is of special interest, since in the latter year tests were made with 58 F water supplied to the cooling coil. The number of days with temperatures above 85 F was 81 for the season of 1936 and 55 for the season of 1935. The numbers of days with temperatures above 90 F were 50 and 19 respectively. The number of hours with temperatures above 85 F for the season of 1936 was 706.3 as compared with 322.6 for 1935, and the numbers of hours with temperatures exceeding 90 F were 341.6 and 82.5 respectively. The severity of the season as measured by degree-hours is apparently determined largely by the total number of days on which the maximum temperature exceeds 90 F, since the higher number of degree-hours is always accompanied by the higher number of hours with temperatures above 90 F. While the 1936 season was severe from the standpoint of outdoor temperatures, it was characterized by much lower outdoor humidities than were obtained for 1935. This is indicated to a certain extent by Fig. 7 from which it may be observed that the outdoor specific humidity ranged from 80 to 100 grains per pound of dry air in 1936 as compared with a range of from 110 to 140 grains per pound for 1935.

From Item 6 in Table 4 it may be observed that the total time that the fan was running during periods when it was circulating air from outdoors at night was 614.4 hours. At the local rate for electrical current of 3.1 cents per kilowatt hour prevailing in Urbana, Ill., the cost of operating the fan during these periods was 1.56 cents per hour. During the stand-by periods from 7 a.m. until the indoor temperature rose to 80 F the fan was allowed to recirculate the air in the house. As shown by Item 10 the total running time for the fan during these periods was 621.0 hours. During all of the time that cooling was demanded, the fan was allowed to run irrespective of whether the water circulating pump was operated or not. The total time that the

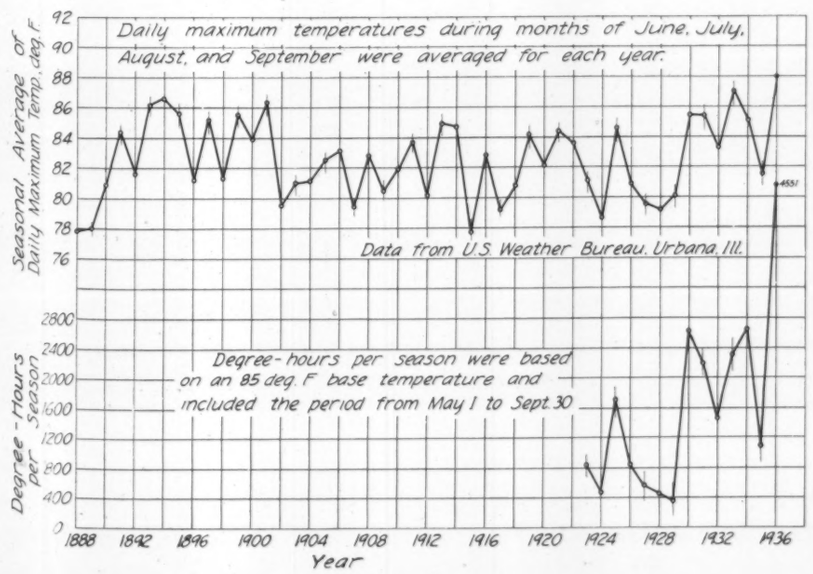

Fig. 9—Graphical record of summer temperatures in Urbana, Ill., from 1888 to 1936

fan was running during these periods, as shown by Item 14, was 695.0 hours. The total time during which the fan was operated to recirculate the air in the house was therefore 1316.0 hours, and the cost of operating the fan to recirculate the air in the house was 1.36 cents per hour.

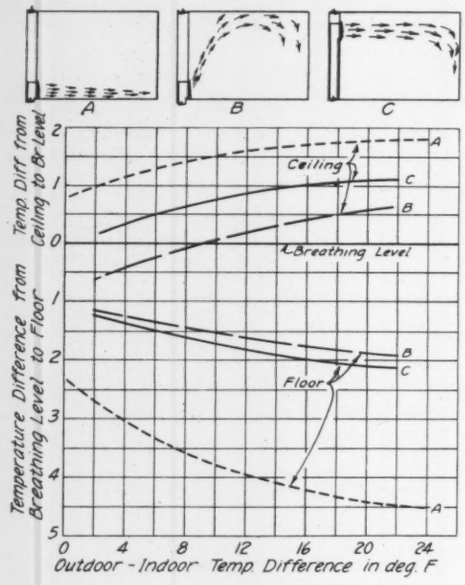

Fig. 10—Room air temperatures with three types of registers

The total amount of water actually used for tests made under conditions of Series 1-36 during the total running time of 211.9 hours was 54,991 gal, and for tests made under conditions of Series 2-36 during the total running time of 240.2 hours was 39,531 gal. At the prevailing rate of 33 cents per thousand gallons, which was also the same as the prevailing rate for the tests of 1935, the hourly costs of water were 8.56 cents and 5.43 cents for Series 1-36 and 2-36 respectively. The total hourly cost for both water and electricity during the actual time when the water was running was 9.9 cents for Series 1-36 and 6.8 cents for Series 2-36, as compared with 13.1 cents for the tests made in 1935. It may be noted that the use of colder water for the cooling coil permitted a marked reduction in the required amount of cooling water, and for equal unit costs for the water, this was reflected in a substantial reduction in the operating costs.

Owing to wide variations in seasonal demands and in local water and electrical rates, seasonal costs are comparatively meaningless. With restriction of due appreciation of their limitations, however, they are undoubtedly of some interest. The first part of the summer from May 1 to June 17 was quite cool and the house temperatures did not attain 81 F at any time during this entire period. Therefore, the actual cooling season extended from June 17, 1936 to September 16, 1936. The total cost for electricity for operating the fan both to circulate air from outdoors at night and to recirculate the air in the house during the day was $27.52. The total cost for water was $31.19 and the cost for both electricity and water was $58.71.

In considering the possible application of cold water as a medium for cooling structures similar to the Research Residence some attention should be given to both the advantages and the disadvantages of the method. These may be itemized as follows:

a. Disadvantages of System

1. The method is confined to those localities in which an abundant supply of cold water is available.

Since water temperatures exceeding 60 F are not suitable, the method is not applicable to those localities supplied with relatively warm surface water. Furthermore, even for localities having a cold water supply, limitations may be imposed as to the quantity of cooling water available for any given installation.

2. High mean coil temperatures are accompanied by high indoor relative humidities. However, the humidity values accompanying mean coil temperatures of approximately 56 F are acceptable.

3. A relatively large amount of coil surface is required. In order to minimize the amount of cooling coil surface correspondingly greater amounts of cooling water must be used.

4. On account of the resistance imposed by the cooling coil to the flow of air, a fan must be selected, which is capable of delivering air against a comparatively high resistance. The use of larger face areas of the cooling coil and the use of lesser number of sections will reduce this air resistance.

5. If the cooling water is supplied from city water mains, the large quantity of water used may materially reduce the water pressure in the service lines to the building. In some cases this may necessitate the use of a larger service line.

b. Advantages of System

1. The system is characterized by simplicity, both of installation and of operation. The controls required for the system are also simple in type and operation.

2. The number of moving parts is few and little servicing is required.

3. The operating costs are reasonably low.

4. The unit is extremely quiet in operation.

5. The water used for cooling may be used for sprinkling the lawn.

6. No additional heat from the operation of the cooling unit is added to the heat load of the house.

Table 5—Temperature Distribution in Four Corners of Room

Arrangement	Description	Deviations in Temperature from Those Obtained at Center of Room[a]									
		Breathing Level					Floor				
		NW	NE	SW	SE	Aver.	NW	NE	SW	SE	Aver.
A	Baseboard, Louvers horizontal	+0.2	—0.5	+0.1	+0.2	0.0	—4.1[b]	—3.1[b]	+1.1	+0.9	—1.3
B	Baseboard, Louvers 60 deg.	—2.0[b]	—0.2	+0.6	+0.6	—0.2	—0.7	—0.2	+0.2	+0.3	0.0
C	Sidewall, Louvers horizontal	—0.4	—1.6[b]	0.0	+0.1	—0.5	+0.4	—0.4	+0.5	+0.6	+0.3

[a] Values preceded by (+) were for temperatures greater than, and values preceded by (—) were for temperatures less than those obtained at the center of the room.
[b] These values were affected to some extent by the cool air from the register.

Register Tests

Three different arrangements of the register were investigated, as illustrated by the insets in the upper portion of Fig. 10. In arrangement *A* the register was located in the baseboard and the louvers were adjusted so that the cool air was projected horizontally into the room near the floor. For arrangement *B* the louvers were set at an angle of 60 deg with the horizontal so that the air was projected upwards into the room. For arrangement *C* the register was located in the sidewall at a height of 7 ft from the floor. In this case the air was delivered horizontally into the upper part of the room.

In Fig. 10 curves are shown giving the relation between the outdoor-indoor temperature difference and the difference in temperatures between the 5 ft level and the floor, and between the ceiling and the 5 ft level. These room air temperatures were obtained by means of

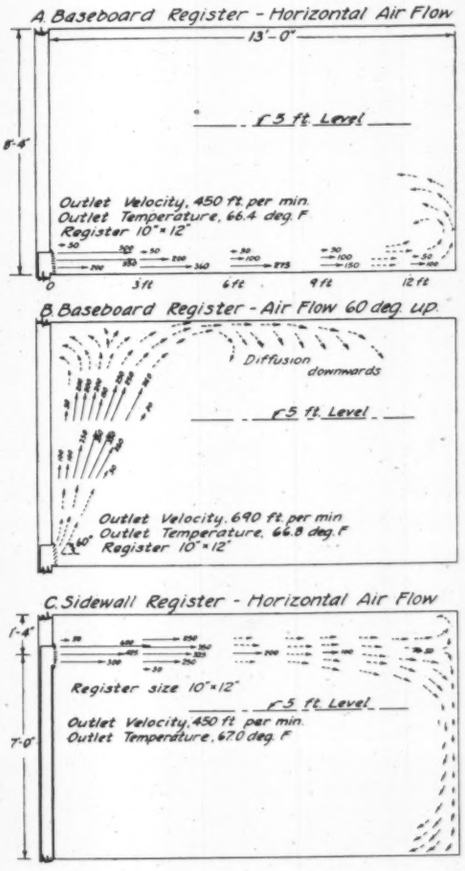

Fig. 11—Diagrams for air circulation in room with three types of register air outlets

thermometers located on the central axis of the room. However, as shown by Table 5, other temperatures observed by means of thermometers located in the four corners of the room indicated that at the 5 ft level the average value of the deviations of these temperatures from those observed in the center of the room was not greater than 0.5 F. It may be noted from Table 5 that some of the temperatures obtained in the north half of the room were affected to some extent by the cool air from the registers. In general, the temperature distribution in all parts of the room was quite uniform, and hence the values obtained in the center may be regarded as typical of those obtained in all parts of the room.

It may be observed from Fig. 10 that in all cases the temperature gradient from the floor to the ceiling tended to increase as the outdoor temperature increased. For an outdoor temperature of approximately 100 F, or for an outdoor-indoor difference of 20 F, the total differences in temperature from the floor to the ceiling were 6.2 F. 2.5 F, and 3.2 F for arrangements *A, B,* and *C* respectively. The temperature gradients obtained for arrangements *B* and *C* were quite small, and even that for arrangement *A* was not excessively large. Furthermore, temperatures in the living zone lower than those in the upper part of the room are not as undesirable in the summer time as they are in the winter. Therefore, it is evident that the performance of registers used for summer cooling work should be evaluated from some standpoint other than that of the temperature gradients produced in the room, and the absence of objectionable air currents or drafts in the living zone is suggested as a better criterion for judging the performance of registers in this case. As indicated by the diagram in Fig. 11, the use of arrangement *A* produced appreciable air currents directly in front of the register, and proved to be unsatisfactory from this standpoint. The air was projected across the floor to the opposite wall at a rather high velocity. This is indicated by the numerical values, for the air velocity in feet per minute, shown in Fig. 11. These values are placed adjacent to the arrows indicating the direction and magnitude of the air flow, and owing to limitations of the instruments used for observing very low velocities, the lower values shown must be regarded as only approximate. The path of the main air stream was such that a considerable portion of the lower part of the room was subjected to objectionable drafts. Although the use of a diffusing type of register, which would tend to distribute the air outwards into the room, might serve to reduce the magnitude of the air velocity, it is probable that objectionable drafts would still be obtained in a considerable zone located in front of the register. Therefore, it may be concluded that a baseboard register with horizontal louvers of the type indicated by arrangement *A* is not satisfactory for summer cooling work.

In the case of arrangement *B* the air flow in the living zone was directed upwards to the ceiling in a path that did not extend beyond approximately 30 in. from the sidewall. Hence, only a very small portion of the living zone was subjected to the direct action of the air from the registers. The cold air after reaching the ceiling diffused and descended into the living zone without the creation of any objectionable drafts. In this connection the arrangement proved to be the most satisfactory of the three tested.

The sidewall register located at the 7 ft level, indicated by arrangement *C* in Fig. 11, projected the air outwards into the room and hence did not create any objectionable drafts in the living zone, except near the outside wall. It was observed that the cool air descended parallel to the outside wall and produced an appreciable draft in a small zone near this wall. Therefore, the only limitation to be observed in connection with the use of arrangement *C* is that the outlet register velocity should

not exceed a value just sufficient to project the air to the wall opposite to the register. Obviously, the use of diffusing type registers, which spread the air outwards into the room, would tend to minimize this effect. But in the selection of such registers due consideration should be given to the proper relations[10] existing between the air velocity, the air temperature, and the size of the room.

Conclusions

The following conclusions may be drawn as applying to the Research Residence and the conditions under which the tests were conducted.

(1) Using water at temperatures of 52 F and 46 F, it is possible to maintain an indoor dry-bulb temperature of approximately 80 F with the outdoor temperature as high as 108 F. On days when the maximum outdoor temperature exceeds 90 F the plant will operate continuously over considerable periods of time and it is possible to maintain an indoor relative humidity of 52 per cent, with a resultant effective temperature of approximately 74 deg, which would undoubtedly be acceptable to the majority of users.

(2) During intermittent operation of the cooling plant the indoor relative humidity rises to a maximum during the *off* period. The magnitude of this rise is dependent to a great extent on the moisture content of the outdoor air.

(3) During continuous operation of the cooling plant the indoor relative humidity decreases to a minimum value which is independent of the moisture content of the outdoor air and is related to the mean coil temperature. Possibly by coincidence the numerical value of the relative humidity that was maintained in the house was nearly the same as the numerical value of the mean coil temperature.

(4) In the case of uninsulated frame structures similar to the Research Residence the more complicated method for calculation of the maximum value of the cooling load, based on including the heat lag, may not be justified.

(5) The absence of objectionable drafts in the living zone was found to be a more satisfactory criterion for evaluating the performance of different types of registers used in connection with summer cooling work than was the existence of small temperature gradients from the floor to the ceiling.

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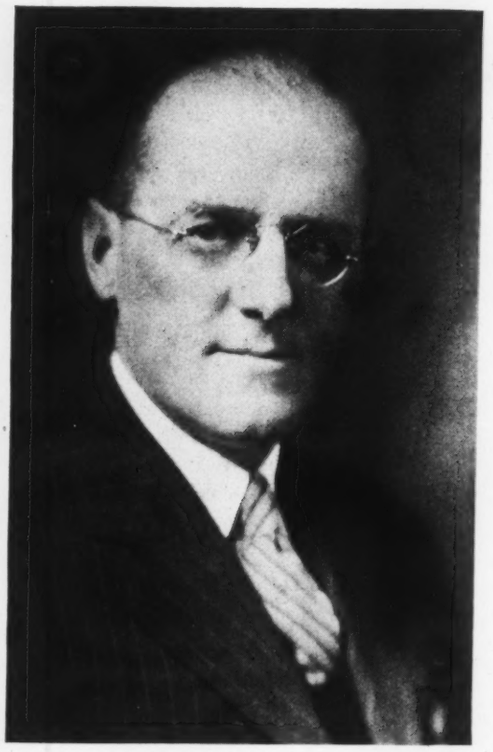
[10]Characteristics of Registers and Grilles, J. H. Van Alsburg, A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, Vol. 7, No. 6, June, 1935, pp. 304-307.

HOSTS FOR JUNE MEETING

The Committee on Arrangements of Massachusetts Chapter Extends a Cordial Welcome to A. S. H. V. E. Members and Offers an Enjoyable Program of Educational and Recreational Features June 24-26



W. T. Jones
General Chairman

A. S. Kellogg
Registration and Reception

James Holt
Entertainment

J. F. Tuttle
Golf

T. F. McCoy
Transportation

W. E. Barnes
Finance

R. M. Nee
Publicity

All Aboard *for* Swampscott

Sea wall in front of New Ocean House

EVERYTHING is in readiness for the Society's Semi-Annual Meeting 1937 which will be held at the New Ocean House, Swampscott, Mass., June 24 to 26. The members of the Massachusetts Chapter have planned a splendid program and hope that Society members will enjoy the hospitality of the New Ocean House, the picturesque scenery and the many historical points of interest in New England.

Swampscott is readily accessible as it is situated twelve miles from Boston on Massachusetts famed North Shore. All of the railroads furnish through car service from leading cities to Boston and excellent connecting trains to Swampscott are provided by the Boston and Maine Railroad.

For those who drive, good roads lead to Swampscott and those who wish to travel by air will find convenient air line schedules to Boston.

The New Ocean House, situated on the water's edge, has a fine private beach and provides every facility for recreation. Nearby points of historic interest are Lexington, Concord, Marblehead, Salem and Plymouth where the Pilgrims landed. Service at the New Ocean House is American Plan and rates include room and meals. Room reservation cards which are being sent to members should be returned promptly to insure the desired type of accommodations.

W. T. Jones, Chairman of the Committee on Arrangements, advises that the Committee has planned entertainment for everyone who attends the Summer Meeting. For those who play golf there will be a Kickers Handicap and a Tournament for the Research and Chapter Cups at the Tedesco Country Club which is situated within a mile of the hotel.

On Thursday afternoon the ladies and members who do not play golf will enjoy a boat trip to Gloucester and in the evening, there will be an informal party for everyone.

Friday morning, the ladies will have a bus trip to Boston and visit the Harvard Museum, the Glass Flower Exhibit, and other points of interest. A special feature for Friday afternoon will be a ladies' bridge and tea and in the evening the Banquet and Dance will be held. The meal will be a typical New England Shore Dinner.

Registration will commence on Wednesday, June 23, when early arrivals will come to Swampscott for Council and Committee Meetings.

Morning Sessions will be held on Thursday, Friday and Saturday at 9:30 a. m. (Daylight Saving Time) and the technical papers will include discussions of air conditioning system operation, the use of pre-cooling coils, performance of fin-tube units, the cooling and heating rates of a room, corrosion studies of steam heating systems, heat requirements of an office building, ventilation requirements and national standards for air conditioning applications.

Members of the Committee on Arrangements assisting Mr. Jones are Alfred S. Kellogg, *Registration and Reception;* James Holt, *Entertainment;* J. F. Tuttle, *Golf;* T. F. McCoy, *Transportation;* R. M. Nee, *Publicity* and W. E. Barnes, *Finance.*

Those members who have visited Swampscott previously have experienced the cordial hospitality of the Massachusetts Chapter members and know the beauties of New England. Those who are making their first visit will find every recreational feature, a comfortable and restful atmosphere, cool sea breezes and a fine opportunity to visit with their fellow members of the A.S.H.V.E.

PLAN YOUR TRIP NOW
Make Hotel Reservations,
Buy Tickets for Train,
Boat, Plane or
Select Your Favorite
Motor Route
COME TO SWAMPSCOTT

PROGRAM

(Daylight Saving Time)

Wednesday, June 23, 1937

1:00 p.m. Registration

2:30 p.m. Council Meeting

6:30 p.m. Guide Publication Committee

8:30 p.m. President's Reception, Music and Dancing

Thursday, June 24, 1937

9:00 a.m. Registration

9:00 a.m. Nominating Committee

9:30 a.m. Technical Papers—

Report of A.S.H.V.E.-A.S.R.E. Committee on National Standards for Air Conditioning Applications, by L. A. Harding, *Chairman*

The Adaptability of Pre-Cooling Coils to Air Washer Systems, by John Everetts Jr.

How Much Power Will an Air Conditioning System Use?, by A. D. Marston

Study of Summer Cooling in the Research Residence Using Water at Temperatures of 52 and 46 F, by A. P. Kratz, S. Konzo and E. L. Broderick

12:30 p.m. Committee on Research

1:30 p.m. Boat Trip to Gloucester

2:00 p.m. Golf Tournament—Tedesco Country Club (18-hole, Kickers Handicap)

8:30 p.m. Informal Party, Concert and Dancing

Friday, June 25, 1937

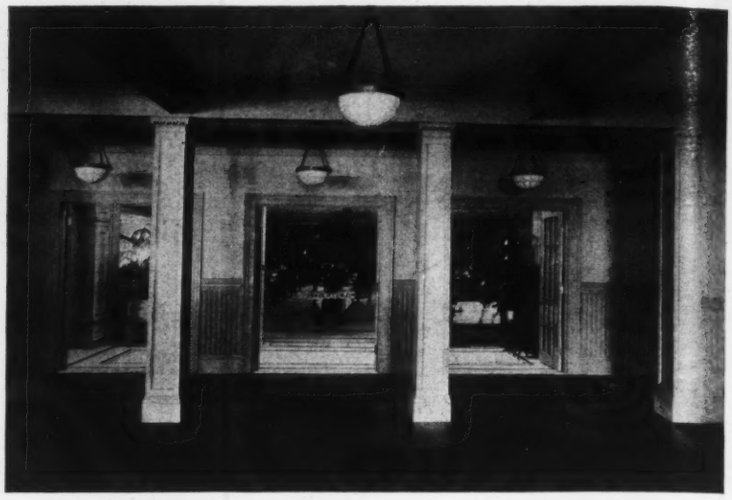
9:30 a.m. Technical Papers—

Calculated Over-all Coefficients for Walls with Air Space Insulation, by F. B. Rowley

Radiation and Convection Across Air Spaces in Frame Construction, by Gordon B. Wilkes and Carl M. F. Peterson

Performance of Fin Tube Units for Air Cooling and Dehumidifying, by G. L. Tuve and C. A. McKeeman

The Cooling and Heating Rates of a Room with Different Types of Steam Radiators and Convectors, by A. P. Kratz, M. K. Fahnestock and E. L. Broderick



Dining room entrance, New Ocean House

10:00 a.m. Bus leaves for Boston—Ladies to visit Harvard Museum and Glass Flower Exhibit

2:00 p.m. Golf Tournament, Tedesco Country Club (18-hole, Medal Play—Research and Chapter Cups)

2:30 p.m. Ladies' Bridge and Tea—Tennis, Archery and other sports

7:30 p.m. Banquet (New England Shore Dinner) and Dance —(Quartette and Speaker)—(Presentation of Past President's Memory Book)

Saturday, June 26, 1937

9:30 a.m. Technical Papers—

An Alternate Method of Comparing the Dust Arrestance of Air Cleaning Devices, by Arthur Nutting

Ventilation Requirements, Part II, by C. P. Yaglou and W. N. Witheridge

Corrosion Studies in Steam Heating Systems, by R. R. Seeber and Margaret R. Holley

Heating Requirements of an Office Building as Influenced by the Stack Effect, by F. C. Houghten and Carl Gutberlet

10:00 a.m. Ladies' visit to North Shore Gardens

Method of Choosing Location of, Financing and Conducting Meetings of the Society

Resolved: That inasmuch as the Annual and Semi-Annual Meetings of the Society come under the jurisdiction of the Council, the following rules governing the handling of such meetings be adopted by the Council and published in the JOURNAL of the Society at least twice during every year, preferably just prior to each meeting:

1—The Council will select the city in which the Annual or Semi-Annual Meeting is to be held, giving due consideration to the invitations received from Chapters or members as well as to the advisability of so distributing those meetings as to make them of the greatest advantage to the general membership, and to reduce as far as possible the expense of members attending.

2—That an appropriation be made to cover the entertainment or local expenses, incurred in connection with the meeting not exceeding $500.00, the regular meeting expense to be taken care of by the General Fund of the Society in the regular way.

3—That no registration fee or compulsory obligations of any nature be imposed on members or guests.

4—That the purchase of tickets for banquets or for any other form of entertainment that may be provided be entirely voluntary.

5—That the grouping of features and the sale of tickets for group features be discouraged.

6—That the raising of funds from manufacturers of heating apparatus be discouraged.

7—That the display of samples, or of literature, advertising the product of any manufacturer in any way, shape or form, be not permitted at the booths, registration desk, or in or about the meetings.

8—That the distribution of trade papers be entirely at the discretion of the committee in charge.

9—That the local Chapter or local members, be empowered to form a General Committee with such sub-committees as may be required to handle the details of transportation, hotel accommodations, entertainment, finance, etc., and that this General Committee be requested to confer frequently with the Council, through the Secretary of the Society, and to make frequent reports on progress in connection with the various matters being handled by them.

10—That the arrangements of elaborate and costly entertainment features be discouraged.

Adopted at Council Meeting, January 29, 1926.

Pittsburgh Experiment Station of the U. S. Bureau of Mines where the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS *is located*

COMMITTEE ON RESEARCH

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Executive Committee: W. A. Danielson, *Chairman;* W. L. Fleisher, J. H. Walker.

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Committee on Air Cleaning—IP-4: H. C. Murphy,* *Chairman;* M. I. Dorfan, C. E. Lewis,* S. R. Lewis, G. W. Penney, A. L. Simison, W. O. Vedder.

Committee on Air Conditioning Requirements of Glass—IF-18: M. L. Carr, *Chairman;* F. L. Bishop, A. N. Finn, E. H. Hobbie, R. J. Lillibridge, R. A. Miller, F. W. Parkinson, W. C. Randall, L. T. Sherwood, J. T. Staples, C. Tasker,* G. B. Watkins, F. C. Weinert.

Committee on Air Distribution—IP-21: Ernest Szekely, *Chairman;* S. H. Downs, M. K. Fahnestock, F. J. Kurth, D. W. Nelson,* C. H. Randolph, J. E. Schoen, G. L. Tuve,* J. H. Van Alsburg.*

Committee on Air Friction—IP-6: J. H. Van Alsburg,* *Chairman;* C. A. Booth, S. H. Downs, C. M. Humphreys, R. D. Madison, L. B. Miller, L. G. Miller.

Committee on Atmospheric Impurities and Resulting Safety and Health Requirements—IP-26: Theodore Hatch, *Chairman;* J. J. Bloomfield, C. A. Booth, Philip Drinker, Dr. Leonard Greenburg, Elliott Harrington,* H. B. Meller.

Committee on Climate and Air Conditioning—C-24: Dr. C. A. Mills, *Chairman;* Major G. C. Dunham, James Govan, Dr. W. J. McConnell, C. P. Neergaard, Dr. F. M. Pottenger, Jr., E. L. Weber, Prof. C.-E. A. Winslow.*

Committee on Comfort Air Conditioning—OH-22: C. Tasker,* *Chairman;* A. E. Beals, F. R. Bichowsky, Thomas Chester, F. E. Giesecke, Elliott Harrington,* Dr. E. V. Hill, R. E. Keyes, C. P. Yaglou.

Committee on Corrosion in Air Conditioning Equipment—IF-14: A. E. Stacey, Jr.,* *Chairman;* A. F. L. Anderson, M. L. Diver, F. L. LaQue, C. E. Lewis,* R. M. Palmer, F. N. Speller, C. M. Sterne, R. T. Thornton, J. H. Young.

Committee on Corrosion in Steam Systems—IF-2: J. H. Walker,* *Chairman;* E. L. Chappell, W. H. Driscoll, C. A. Dunham,* L. B. Miller, R. R. Seeber, C. M. Sterne.

Committee on General Air Conditioning Comfort Requirements—OH-5: C. P. Yaglou, *Chairman;* J. J. Aeberly, R. R. Sayers, C.-E. A. Winslow.*

Committee on Heat Requirements of Buildings—IF-8: O. W Armspach, *Chairman;* P. D. Close, W. H. Driscoll, H. M. Hart, V. W. Hunter, H. H. Mather, E. C. Rack, F. B. Rowley, R. J. J. Tennant, J. H. Walker.*

Committee on Heat Transfer of Finned Tubes with Forced Air Circulation—IP-10: F. B. Rowley, *Chairman;* H. F. Hutzel, R. H. Norris, C. H. Randolph, W. E. Stark, G. L. Tuve,* C. F. Wood.

Committee on Insulation—IF-23: L. A. Harding, *Chairman;* E. A. Allcut, J. D. Edwards, E. C. Lloyd, W. E. McMullen, R. T. Miller, E. R. Queer, T. S. Rogers, F. B. Rowley, W. S. Steele, C. Tasker,* B. Townshend, G. B. Wilkes.

Committee on Intermittent Heating—IP-20: E. K. Campbell, *Chairman;* W. L. Cassell, Prof. E. F. Dawson, N. W. Downes, F. E. Giesecke, J. M. Robertson, J. H. Kitchen, Prof. A. H. Sluss, G. L. Tuve.*

Committee on Psychrometry—C-11: F. R. Bichowsky, *Chairman;* C. A. Bulkeley, J. A. Goff, Dr. E. V. Hill, F. G. Keyes, A. P. Kratz,* W. M. Sawdon.

Committee on Radiation with Gravity Air Circulation—IP-9: M. K. Fahnestock, *Chairman;* H. F. Hutzel, J. P. Magos, J. W. McElgin, J. F. McIntire, D. W. Nelson,* R. N. Trane, T. A. Novotney.

Committee on Relation of Body Changes to Air Changes—OH-3: Dr. E. V. Hill, *Chairman;* N. D. Adams, J. J. Aeberly, John Howatt, A. P. Kratz,* P. J. Marschall, V. L. Sherman.

Committee on Sound Control—IF-1: J. S. Parkinson, *Chairman;* C. M. Ashley, V. O. Knudsen, R. F. Norris, C. H. Randolph, J. P. Reis, A. E. Stacey,* F. R. Watson.

Committee on Summer Air Conditioning for Residences—IP-7: M. K. Fahnestock, *Chairman;* E. A. Brandt, John Everetts. Jr., Elliott Harrington,* H. F. Hutzel, E. D. Milener, K. W. Miller, E. B. Newill, F. G. Sedgwick, J. H. Walker.*

Committee on Treatment of Air with Electricity—C-17: Prof. C.-E. A. Winslow,* *Chairman;* R. D. Bennett, W. H. Carrier, L. W. Chubb, Major W. D. Fleming, R. F. James, L. R. Koller, Dr. C. A. Mills, Prof. E. B. Phelps, Prof. G. R. Wait, Prof. W. T. Wells.

*Member of Committee on Research.

NEWS OF LOCAL CHAPTERS

OFFICERS OF LOCAL CHAPTERS—1937

ATLANTA: *Organized,* 1937. *Headquarters,* Atlanta, Ga. *President,* E. W. KLEIN, 152 Nassau St., N. W. *Secretary,* C. T. BAKER, 713 Glenn St., S. W.

CINCINNATI: *Organized,* 1932. *Headquarters,* Cincinnati, O. *Meets,* Second Tuesday in Month. *President,* I. B. HELBURN, 610 Chamber of Commerce Bldg. *Secretary,* H. E. SPROULL, 1005 American Bldg.

ILLINOIS: *Organized* 1906. *Headquarters,* Chicago, Ill. *Meets,* Second Monday. *President,* L. S. RIES, 960 East 58th St. *Secretary,* C. E. PRICE, 6 N. Michigan Ave.

KANSAS CITY: *Organized,* 1917. *Headquarters,* Kansas City, Mo. *Meets,* Second Monday in Month. *President,* L. R. CHASE, 2440 Pennway. *Secretary,* F. J. DEAN, JR., 6028 Walnut St.

MANITOBA: *Organized,* 1935. *Headquarters,* Winnipeg, Man. *Meets,* Fourth Thursday. *President,* J. B. STEELE, 184 Waterloo St. *Secretary,* C. H. TURLAND, 325 Centennial St.

MASSACHUSETTS: *Organized,* 1912. *Headquarters,* Boston, Mass. *Meets,* Second Tuesday in Month. *President,* JAMES HOLT, Massachusetts Institute of Technology, Cambridge, Mass. *Secretary,* JOHN TURNER, 285 Columbus Ave.

MICHIGAN: *Organized,* 1916. *Headquarters,* Detroit, Mich. *Meets,* First Monday after the 10th of the Month. *President,* R. K. MILWARD, 127 Campbell Ave. *Secretary,* G. H. TUTTLE, 2000 Second Ave.

WESTERN MICHIGAN: *Organized,* 1931. *Headquarters,* Grand Rapids, Mich. *Meets,* Second Monday in Month. *President,* L. G. MILLER, Mich. State College, E. Lansing, Mich. *Secretary,* E. I. ADAMS, 115 So. Pine, Lansing, Mich.

MINNESOTA: *Organized,* 1918. *Headquarters,* Minneapolis, Minn. *Meets,* Second Monday in Month. *President,* N. D. ADAMS, 220 Second Ave. S. W., Rochester, Minn. *Secretary,* J. E. SWENSON, 800 Hennepin Ave.

MONTREAL: *Organized,* 1936. *Headquarters,* Montreal, Que. *Meets,* Third Monday. *President,* G. L. WIGGS, University Tower. *Secretary,* C. W. JOHNSON, 630 Dorchester St., W.

NEW YORK: *Organized,* 1911. *Headquarters,* New York, N. Y. *Meets,* Third Monday in Month. *President,* G. E. OLSEN, 101 Park Ave. *Secretary,* T. W. REYNOLDS, 100 Pinecrest Dr., Hastings-on-Hudson, N. Y.

WESTERN NEW YORK: *Organized,* 1919. *Headquarters,* Buffalo, N. Y. *Meets,* Second Monday in Month. *President,* P. S. HEDLEY, Curtiss Bldg. *Secretary,* C. A. GIFFORD, 758 Parkside Ave.

NORTHERN OHIO: *Organized,* 1916. *Headquarters,* Cleveland, O. *Meets,* Second Thursday in Month. *President,* PHILIP COHEN, 401 East Ohio Gas Bldg. *Secretary,* C. A. MCKEEMAN, Case School of Applied Science.

OKLAHOMA: *Organized,* 1935. *Headquarters,* Oklahoma City, Okla. *Meets,* Second Monday. President, F. X. LOEFFLER, 1604 N. W. Fifth St. *Secretary,* E. W. GRAY, Box 1498.

ONTARIO: *Organized,* 1922. *Headquarters,* Toronto, Ont. *Meets,* First Monday in Month. *President,* G. A. PLAYFAIR, 97 Jarvis St. *Secretary,* H. R. ROTH, 57 Bloor St. W.

PACIFIC NORTHWEST: *Organized,* 1928. *Headquarters,* Seattle, Wash. *Meets,* Second Tuesday in Month. *President,* S. D. PETERSON, 473 Colman Bldg. *Secretary,* D. C. GRIFFIN, Orpheum Bldg.

PHILADELPHIA: *Organized,* 1916. *Headquarters,* Philadelphia, Pa. *Meets,* Second Thursday in Month. *President,* L. P. HYNES, 240 Cherry St. *Secretary,* C. B. EASTMAN, 530 Brookview Lane, Brookline, Upper Darby, Pa.

PITTSBURGH: *Organized,* 1919. *Headquarters,* Pittsburgh, Pa. *Meets,* Second Monday in Month. *President,* M. L. CARR, P. O. Box 1646. *Secretary,* T. F. ROCKWELL, Carnegie Inst. Tech.

ST. LOUIS: *Organized,* 1918. *Headquarters,* St. Louis, Mo. *Meets,* First Tuesday in Month. *President,* G. W. F. MYERS, 3947 W. Pine Blvd. *Secretary,* D. J. FAGIN, 1344 Woodruff Ave.

SOUTHERN CALIFORNIA: *Organized,* 1930. *Headquarters,* Los Angeles, Calif. *Meets,* Second Tuesday in Month. *President,* E. H. KENDALL, 1978 S. Los Angeles St. *Secretary,* J. F. PARK, 1234 South Grand.

TEXAS: *Organized,* 1936. *Headquarters,* College Station, Texas. *President,* F. E. GIESECKE, Texas A. & M. College. *Secretary,* W. H. BADGETT, Texas A. & M. College.

WASHINGTON, D. C.: *Organized,* 1935. *Headquarters,* Washington, D. C. *Meets,* Second Wednesday in Month. *President,* W. A. DANIELSON, 2236 Munitions Bldg. *Secretary,* M. D. KICZALES, 3000 Connecticut Ave., N. W.

WISCONSIN: *Organized,* 1922. *Headquarters,* Milwaukee, Wis. *Meets,* Third Monday in Month. *President,* M. E. ERICKSON, 8405 W. Greenfield Ave., West Allis, Wis. *Secretary,* R. G. KOCK, 626 E. Wisconsin Ave.

Atlanta Chapter Elects Officers

April 22, 1937. Atlanta members of the A.S.H.V.E. held an organization meeting for the purpose of establishing a local chapter and electing officers. The meeting was called to order by the temporary chairman C. T. Baker who read a telegram from D. S. Boyden, president of the Society, congratulating Atlanta members on the perfecting of a chapter organization and advising them that he would come to Atlanta at some future date to present the charter.

This was followed by the reading of the report of the Nominating Committee composed of W. J. McKinney, chairman, L. Sudderth, Jr., and Roy Kahn, recommending the following members as officers:

President—E. W. Klein
Vice-President—T. T. Tucker
Secretary-Treasurer—C. T. Baker
Board of Governors—F. W. Clare, W. H. Moler

Following the reading of this report, ballots were distributed and a vote taken which resulted in the nominees being elected to the offices as recommended by the Nominating Committee.

The gavel was then turned over to Pres. E. W. Klein, who gave a short talk and welcomed the non-members present. President Klein called on Secretary Baker to read a number of telegrams and letters received from many of the Chapters throughout the United States and Canada congratulating Atlanta members.

The next order of business was the reading of a resolution which was adopted by the Society's Council favoring passage of House Resolution No. 3629 providing for the establishment of engineering experiment stations at land grant colleges. The merits of this resolution were recognized and Secretary Baker was instructed to send copies to the U. S. Senators and Representatives from the district, with a letter requesting that the resolution be given serious consideration when brought up for discussion in the House of Representatives.

L. F. Kent was appointed by President Klein to be in charge of newspaper publicity in connection with Society activities.

President Klein then announced a round table discussion for the "Good of the Order" and this was followed by a Dutch supper, the details of which had been arranged by Vice Pres. T. T. Tucker.

Coil Steam Generating Unit Described at Illinois Meeting

April 12, 1937. Pres. J. J. Hayes presided at the regular meeting of the Illinois Chapter held in the West Room of the Hotel Sherman with 78 present for the dinner and meeting.

The minutes of the March meeting were read by Secy. C. E. Price as well as several other communications having to do with employment. An announcement was made that the meeting on May 10 would be honored by the presence of D. S. Boyden, president of the A.S.H.V.E.

The following members were accepted into the Chapter and their applications were read by Tom Brown, chairman of the Membership Committee: C. M. Burnam, Jr., B. C. Benson and D. C. Wiley.

The subject of the paper presented was the Clarkson Coil Steam Generating Unit. President Hayes introduced E. A. Russell, Chicago, chief engineer, Vapor Car Heating Co., who explained that with the new types of diesel-powered railroad trains it is necessary to install a boiler to supply steam for heating. It is important that the weight of the heating equipment be as little as possible and this has led to the development of the coil steam generating unit.

Mr. Russell then turned the discussion over to Alick Clarkson, consulting engineer, Vapor Car Heating Co., who presented a detailed description of the unit which he had developed. Among the problems that had to be solved were: control; housing—one that will not be destroyed and will allow for replacement of coils as required; coils that will not sag. Following a description of early flash steam boilers, illustrated with slides, the speaker explained how the unit is controlled and how the proportion of fuel to water is governed and he also gave heat release and efficiency figures. It was mentioned that such a steam generating unit must be able to burn any oil that the diesels will operate on.

In the general discussion questions were asked on the cost of the unit and how much noise it causes. These were answered by Mr. Clarkson and Mr. Russell following which they invited anyone interested to visit their testing laboratory to see the unit in operation.

After a rising vote of thanks to the speakers the meeting adjourned at 9:00 p.m.

Information on Glass Block Presented to Wisconsin Chapter

April 21, 1937. The meeting of Wisconsin Chapter was called to order at 8:00 p.m. and Secy. R. G. Koch was instructed to send a congratulatory telegram to the new A.S.H.V.E. chapter at Atlanta, Ga.

It was further resolved that Secretary Koch should write to the congressional representative urging passage of House Resolution No. 3629 relating to the establishment of experiment engineering stations at land grant colleges.

Following the business meeting E. P. Lockart, Owens-Illinois Glass Co., gave technical information of interest in a talk on Glass Block. Mr. Lockart also presented some moving pictures showing manufacturing processes and various tests, as well as views of different applications of glass block in building.

Texas Chapter Receives Charter from President Boyden and Elects Officers

April 15, 1937. The April meeting of Texas Chapter was held in the Adolphus Hotel, Dallas, with Prof. F. E. Giesecke, president, presiding. There were 26 members and guests at the meeting which was called to order at 7:30 p.m., and which had been preceded by a dinner earlier in the evening honoring D. S. Boyden, president of the A.S.H.V.E.

President Boyden presented the Texas Chapter with its charter, which was accepted on behalf of the chapter by Professor Giesecke.

The minutes of the March 20 meeting, which was held at Houston, were read by Prof. E. G. Smith in the absence of the secretary and were approved.

President Giesecke explained the method of nominating and electing officers as prescribed in the Chapter By-Laws and requested R. K. Werner, chairman of the Nominating Committee, to present the report of his committee on the nomination of officers to serve for the year beginning May 1937. Nominees selected by the committee were announced as follows:

President—R. F. Taylor
Vice-President—H. W. Skinner
Secretary—W. H. Badgett
Treasurer—I. E. Rowe
Board of Governors—J. A. Kiesling, M. L. Diver, Charles Kribs

President Boyden was called upon to give a resumé of Society activities and he spoke briefly on high points of the Society's history and accomplishments and outlined research work in air conditioning now being undertaken by the A.S.H.V.E. W. H. Carrier, Newark, N J., past president of the Society, was present and spoke briefly on the history of air conditioning.

Following Mr. Carrier's talk, President Boyden spoke on the subject of The Economic Use of Purchased Steam, in which he explained unusual aspects of the growing industry which supplies steam from a central generating source for the heating of apartment and office buildings.

After the discussion of President Boyden's address, Mr. Carrier gave an interesting description of air conditioning in the gold mines of South Africa.

A discussion of the New London disaster took place and it was voted that the Governor be requested to appoint representatives of the Texas Chapter of the A.S.H.V.E. on the committee to compile a uniform building code for the state.

R. F. Taylor presented President Boyden with an invitation to attend the Pan-American Exposition to be held at Dallas during the summer.

The meeting adourned at 11:00 p.m. with the announcement that Chapter members would meet at College Station in May.

Professor Kratz Speaks at Joint Meeting in St. Louis

March 4, 1937. Pres. C. R. Davis called the St. Louis Chapter meeting to order at 7:20 p.m. at Garavelli's Restaurant, and among the 18 members and guests present was Prof. A. P. Kratz, University of Illinois.

Secy. R. J. Tenkonohy reported that J. J. Blackmore had affiliated with the Chapter and that the membership application of C. V. Allen had been forwarded to the headquarters office.

C. E. Hartwein gave the treasurer's report, and D. J. Fagin of the Program Committee stated that Pres. D. S. Boyden was scheduled as speaker for the April 12 meeting.

G. W. F. Myers of the Membership Committee announced that applications had been received from C. A. Pickett and J. A. Badaracco.

President Davis requested L. R. Szombathy, chairman of the Entertainment Committee, to make plans for the June party.

J. W. Cooper, P. W. Sodemann and L. W. Moon were elected to the nominating Committee, and President Davis appointed E. A. White and J. M. Foster, and named Mr. Moon as chairman.

A recess was declared and the meeting reconvened at the Engineers Club for the technical program, with 180 members and guests present. President Kraft, Engineers Club, presided and stated that the occasion was a joint meeting of the Engineers Club, the St. Louis Chapter of the A.S.H.V.E. and the local section of the *A.S.R.E.*, and he called upon President Davis to introduce the speaker, Professor Kratz.

Professor Kratz gave a very interesting presentation of data on the Cooling of the Research Residence at the University of Illinois, and illustrated his talk with slides showing photographs and charts. The work covered tests carried on over a number of years with various methods of cooling, and after closing his formal talk, Professor Kratz replied to the questions in the discussion which followed.

Pittsburgh Chapter Has E. H. Gurney as Guest Speaker

April 19, 1937. Pres. M. L. Carr presided at the Pittsburgh Chapter meeting held at the downtown Y. M. C. A. and Secy. T. F. Rockwell reported the minutes of the March meeting which were approved.

Reports were in order and Treas. J. E. McLean gave a statement on the chapter's financial condition, and E. C. Smyers of the Program Committee announced that the May meeting would be devoted to a symposium on atmospheric pollution. P. A. Edwards reported that a total of nine applications for membership in the Society had been forwarded to headquarters office since the beginning of the year.

Secretary Rockwell read a resolution recently adopted by the

A.S.H.V.E. Council concerning federal aid for engineering experiment stations at land grant colleges.

It was also announced that a charter had been granted to a chapter in Atlanta, Ga.

Before proceeding with the regular program, President Carr introduced R. J. Tenkonohy, Secretary of St. Louis Chapter, who spoke briefly on the experiences of the St. Louis group during the Annual Meeting in January.

J. V. Clippenger, senior student in heating and ventilating at Carnegie Institute of Technology, was introduced and abstracted a report on the operating experiences of the air conditioning system in the Philadelphia Savings Fund Building.

E. H. Gurney, Toronto, Ont., first vice-president of the A.S.H.V.E., was the principal speaker of the evening and was presented by President Carr. Mr. Gurney, who had recently returned from a trip to the Union of South Africa, chose to speak on some of the things he saw during his travels. He described how the current boom in gold mining in South Africa has created a new demand for air conditioning in the great need for mechanical cooling at the working faces below the 5,000 ft level. Mr. Gurney also predicted a growing market for other conditioning equipment in the cities of South Africa, especially in the field of dust removal. Concluding the description of his trip, Mr. Gurney asked that the Chapter help to maintain the high ethical standards of the A.S.H.V.E. and urged continued support of the Society's research work.

The meeting adjourned at 10:00 p.m. with 48 members and guests present.

Pres. D. S. Boyden Guest and Speaker at Southern California Meeting

April 20, 1937. The April meeting of Southern California Chapter was held at the Clark Hotel on the occasion of Pres. D. S. Boyden's visit to Los Angeles. This was a joint meeting with the local section of the *American Society of Refrigerating Engineers* and 91 members and guests were present to welcome Colonel Boyden, president of the A.S.H.V.E. Following dinner entertainment by Mr. and Mrs. Itinkigohome, Japanese magicians, was enjoyed by the members and guests.

Pres. Leo Hungerford presided and called upon Secy. H. M. Hendrickson to read a telegram regarding the June meeting of the A.S.H.V.E. at Swampscott, Mass.

E. H. Kendall, chairman of the Program Committee, announced the May meeting to be a joint gathering with the *A.S.R.E.*, with Prof. B. M. Woods, dean of mechanical engineering at the University of California, Berkeley, as the principal speaker. It was planned to have Professor Woods talk on the subject of Cooling Towers and Surface Cooling.

President Hungerford welcomed the *A.S.R.E.* members and guests and then presented Chairman J. C. Blair of *A.S.R.E.*, who expressed the pleasure of his group at being present.

W. E. Barnum, chairman of the Nominating Committee, reported that the Nominating Committee consisting of L. H. Polderman, A. J. Hess, A. G. Orear and W. E. Barnum proposed the following slate of officers for the coming year:

President—E. H. Kendall
Vice-President—H. M. Hendrickson
Secretary—J. F. Park
Treasurer—A. W. Cooper
Board of Governors—O. W. Ott, P. C. Scofield, W. D. Fabling, Leo Hungerford

It was announced that ballots would be sent out by mail before the next meeting so that the vote might be counted in May.

In introducing Colonel Boyden, Mr. Hungerford told of the new president's work for the A.S.H.V.E. and urged all those present who were engaged in heating, ventilating and air conditioning to become members of the Society, if they had not already done so.

President Boyden devoted the first part of his address to the history of the Society's growth and a review of its activities, and gave a statement of its present size and financial position.

The remainder of President Boyden's talk was on the subject of the Economic Use of Purchased Steam which covered the various uses of district steam for heating buildings, controls for such heating service and condensate reclaiming for use as domestic hot water. The talk was illustrated with slides and proved to be very interesting as well as instructive.

President Hungerford thanked Colonel Boyden for his fine address and expressed the pleasure of the local chapter on having him as their guest, before the meeting adjourned at 9:30 p.m.

Northern Ohio Chapter Enjoys Talk by E. H. Gurney

April 21, 1937. Approximately 45 members and guests attended the Northern Ohio Chapter meeting which was preceded by a dinner when 28 members were present.

Minutes of the previous meeting were read and approved and R. A. Wilson reported the membership application of Walter Bagly, who was unanimously accepted as a Chapter member.

F. A. Kitchen reported on the suggested changes in the By-Laws and it was moved and seconded that the revised By-Laws be accepted and that the new copies be distributed.

Ohio State House Bill No. 623 was discussed and a resolution was moved and seconded that the Chapter express disapproval of this bill to the Senate Labor Committee.

Philip Cohen, H. E. Wetzel, and J. G. Harris were appointed by Pres. L. T. Avery as members of the Auditing Committee.

Nominations were received from the floor and resulted in the election of Messrs. Kitchen, Avery and Wilson to serve as the Nominating Committee and choose prospective officers to be elected at the next Chapter meeting.

E. H. Gurney, Toronto, Ont., first vice-president of the A.S.H.V.E. was the guest speaker of the evening and was introduced by C. F. Eveleth. Mr. Gurney gave a very interesting talk on gold mining and its problems in Johannesburg, South Africa, and related that air conditioning is beginning to play a part in this work, as the control of temperatures and dust are important factors. A general discussion followed regarding A.S.H.V.E. activities, in which many took part

According to the report of Secy. W. R. Beach the meeting adjourned at 10:30 p. m.

Pres. Boyden and Council Members Attend Washington Meeting

April 9, 1937. Washington, D. C., Chapter met at Wesley Hall and the group was honored by the attendance of A. S. H. V. E. Council members, who had held a meeting during the afternoon in the Mayflower Hotel. Dinner was served at 7:30 p. m. and there were approximately 65 members and guests present, including the following officers of the Society and members of the Council: Pres. D. S. Boyden, Boston, First Vice-Pres. E. H. Gurney, Toronto, Treas. A. J. Offner, New York, F. E. Giesecke, College Station, Tex., W. E. Stark, Cleveland, G. L. Larson, Madison, Wis., R. C. Bolsinger, Conshohocken, Pa., S. H. Downs, Kalamazoo, J. J. Aeberly, Chicago, M. C. Beman, Buffalo, W. A. Russell, Kansas City, and A. V. Hutchinson, New York, secretary. Also present at the dinner were F. C. Houghten, director of the Society's Research Laboratory, Pittsburgh, Pa., J. D. Cassell and M. F. Blankin of Philadelphia, and Messrs. Hanline and Meigs, Washington, D. C. J. F. McIntire, Detroit, second vice-president of the A. S. H. V. E., and W. L. Fleisher, New York, member of the Council, were present at the meeting later in the evening.

Pres. W. A. Danielson called the assembly to order at 8:45 p.m. and the minutes of the preceding meeting were read and approved with correction.

The Nominating Committee reported the following nominations for officers to serve the Chapter for the season of 1937-1938:

President—L. S. Ourusoff.
Vice-President—E. J. Febrey.
Secretary—L. F. Nordine.
Treasurer—Glegge Thomas.
Board of Governors—T. H. Urdahl, W. E. Kingswell, M. D. Kiczales.

Mr. Febrey asked that his name be withdrawn and the Nominating Committee thereupon reported the name of S. P. Eagleton for the office of vice-president.

It was the recommendation of the members of the Nominating Committee that President Danielson should be tendered the title of honorary president of the Chapter, in token of appreciation of his excellent service.

It was also recommended by E. V. Fineran and C. S. Stock of the Nominating Committee that R. H. Feltwell, chairman of the Nominating Committee, be tendered the title of honorary president in view of his excellent work both in the formation of the Chapter and his interest and activity subsequent thereto.

It was announced that the May meeting was to be held in the new Interior Building and that a talk was to be given on Centrifugal Compressors.

On motion duly made and seconded, a resolution was unanimously adopted urging passage of House Resolution 3629. It was suggested that a copy of the resolution be sent to each member of the Chapter and that the members be urged to communicate with legislators and others in an effort to secure the passage of the bill.

Chairman Danielson presented Pres. D. S. Boyden, Boston, president of A. S. H. V. E., who introduced each member of the Council present. President Boyden then outlined the history of the Society, showing how the founders formulated certain policies based upon predictions for the future of the industry, which up to date have been realized. He also spoke of the Society's research work and stressed the value to the national Society of the local Chapters.

President Boyden reported that an Exposition Committee had been appointed in connection with the Heating and Ventilating Show to be held in New York in January, 1938.

President Danielson then introduced E. H. Gurney, first vice-president of the A. S. H. V. E., who gave a very entertaining talk on his experiences in South Africa, during which he mentioned the possible installation of a dust removal system in Johannesburg.

President Danielson, as chairman of the Research Committee, gave a short resumé of the work being carried on by the committee.

There being no further business to come before the meeting, the session adjourned at 10:30 p. m.

Advisory Committee Appointed for Fifth Exposition

Announcement has been made of the appointment of the following members of the Advisory Committee of the Fifth International Heating and Ventilating Exposition: *Chairman,* D. S. Boyden, Edison Electric Illuminating Co., Boston; *Vice-Chairman,* E. H. Gurney, Gurney Foundry Co., Ltd., Toronto; E. E. Ashley, consulting engineer, New York, N. Y.; Potter Bowles, Hoffman Specialty Co., New York; W. H. Carrier, Carrier Corp., Newark; Lt. Col. W. A. Danielson, Washington; W. L. Fleisher, consulting engineer, New York; A. R. Herske, American Radiator Co., New York; Prof. G. L. Larson, University of Wisconsin, Madison, Wis.; A. J. Offner, consulting engineer, New York; C. R. Place, consulting engineer, New York; D. J. Purinton, Mahony-Troast Construction Co., New York; L. L. Jacobs, *Oil Burner Institute,* Clifton, N. J.; H. T. Richardson, *National Warm Air Heating and Air Conditioning Association,* New York; E. W. Smith, *Institute of Boiler and Radiator Manufacturers,* Westfield, Mass.; H. P. Morehouse, *American Gas Association,* Newark; Homer Addams, *Steel Heating Boiler Institute,* New York; and G. H. Dickerson, *Heating, Piping and Air Conditioning Contractors National Association,* Chicago, Ill.

The Exposition will be held in New York, N. Y., during the week of January 24, 1938, coincident with the 44th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. C. F. Roth, Grand Central Palace, New York, N. Y., who has been similarly responsible for previous expositions, is in charge of arrangements.

Institute on Coal Utilization

The First Institute on Coal Utilization was held at the University of Michigan, Ann Arbor, Mich., April 13, 14 and 15, and was well attended by Michigan fuel, stoker and heating dealers.

Discussions were given by Irvin Davis, Hatfield-Campbell Creek Coal Co., Cincinnati, on Merchandising, J. E. Tobey, fuel engineering division, Appalachian Coals, Inc., who spoke on the fuel engineering field, and G. W. McCready, assistant professor of mechanical engineering, who analyzed heating values of coal, gas and oil as fuel.

The development and use of the smaller stoker was covered by R. S. Hawley, professor of mechanical engineering. Automatic control of heating devices was discussed by E. K. Scoggin, Minneapolis-Honeywell Regulator Co. The use of Pocahontas coal in stokers was treated analytically by R. M. Pilcher, Norfolk & Western Railway.

M. G. Bluth, Committee of Ten, spoke on the use of coal in Michigan's old and new homes, and automatic heating and air conditioning were discussed by Axel Marin, associate professor of mechanical engineering. A paper on insulation was presented by Douglas Via, Eagle-Picher Sales Co., Cleveland. Future heating from a research angle was the subject of Prof. C. W. Good's talk and A. T. Murphy, of the Black Diamond, spoke on the topic of Sit Down Program for the Retail Coal Merchants. D. H. Neff, managing director, *Michigan Retail Coal Merchant's Association,* discussed the retail coal dealer's problem in the State of Michigan.

Heating and Air Conditioning Conference at Iowa State

Iowa State College, Ames, Iowa, held its Second Annual Heating and Air Conditioning Conference, April 7, 8, and 9, with sessions which were attended by architects, builders, heating contractors and dealers, engineers, coal, stoker heating and air conditioning dealers, and stoker manufacturers.

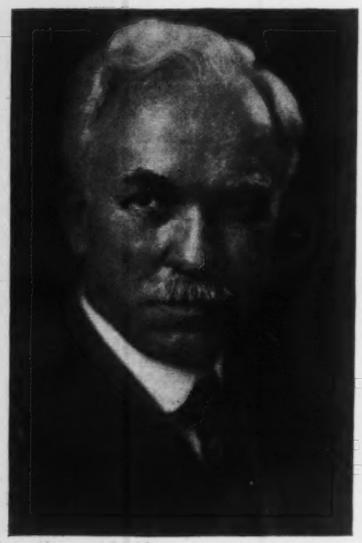
Scheduled on the three-day program were Byron Eaton, Delco-Frigidaire Corp., who discussed developments in heating and the sale of comfort; Elliott Harrington, General Electric Co., who covered 1937 trends in air conditioning; and S. Konzo, University of Illinois, who described summer air conditioning in residences.

Other speakers were T. C. Russell, Russell Electric Co., who spoke on fans in relation to heating and cooling systems; B. E. Shaw, Penn Electric Switch Co., whose subject was automatic control of temperature and humidity; and C. W. Nessle, Minneapolis-Honeywell Regulator Co., on controls for stokers and oil burners. Suggestions for the efficient operation of small stokers were made by R. E. Young, Whiting Corp., and activities in the stoker industry were discussed by K. C. Richmond, Coal-Heat. A. C. Walters, Green Furnace & Foundry Works, discussed figuring a forced warm air heating system, and F. L. Myers, Owens-Illinois Glass Co., spoke on air filters and their use.

Problems covering the requirements and preparation of stoker coal were explained by H. A. Christopherson, Iowa Coals, Inc., and results of a number of tests on Iowa coal in small stokers were given by R. J. Helfinstine, Iowa State College. Insulation as it affects heating and air conditioning was discussed by Russell Backstrom, Wood Conversion Co., and L. V. Teesdale, U. S. Forest Products Laboratory, discussed moisture accumulation in walls and attic spaces.

Dr. O. P. Hood, Honorary Member, Dies

In the death of Dr. O. P. Hood the A.S.H.V.E. has lost an Honorary Member whose valued support and loyalty could always be counted upon to extend the interests of the Society. Dr. Hood, who was 71 years of age and until recently chief of the technological branch of the U. S. Bureau of Mines, died April 22, 1937 at his residence, 1831 Irving St., Washington, D. C., after a long illness. He was retired June 30, 1936, following 25 years of service with the Bureau, after his term of office had been extended one year beyond the age limit by executive order.



O. P. Hood

Ozni Porter Hood, the son of an inventor, was born in Lowell, Mass., on June 14, 1865, was educated in mechanical engineering in Worcester and Rose Polytechnic Institutes, and served as professor of mechanical engineering, first at Kansas Agricultural College and then at the Michigan College of Mines.

Dr. Hood joined the Bureau of Mines at its Pittsburgh Experiment Station in 1911. In addition to the many administrative duties of the chief of a major division in the Bureau, Dr. Hood found time to plan and direct the design of the new building for the Pittsburgh Experiment Station. This building, completed in 1918, remains today an outstanding tribute to his genius in carrying out the wishes of Dr. J. A. Holmes, first director of the Bureau.

Since 1919 when the A.S.H.V.E. Research Laboratory was established it has been housed in this Bureau of Mines Building and the cooperative research work that has been carried on had Dr. Hood's active encouragement and support as well as his unfailing interest.

Following the completion of the station in 1918, Dr. Hood was called to the Washington Office in order to meet widespread demands on the Bureau in the field of fuels and mechanical equipment.

In 1926 Dr. Hood again responded to the needs of the Bureau in accepting the additional duties of Chief of the Technological Branch. To this position he brought a wealth of experience and a maturity of judgment which helped carry the Bureau through a trying period.

His retirement completed a quarter century of service in which he helped to guide the Bureau of Mines from infancy to maturity. His fine administrative record, as well as his technical achievements, won for him a high position in his profession. He was made an Honorary Member of the American Society of Heating and Ventilating Engineers in 1929, and served as ex-officio member of the Society's Committee on Research. He was an Honorary Member of the *American Society of Mechanical Engineers*, the *American Institute of Mining and Metallurgical Engineers*, and the *International Railway Fuel Association*, and in 1932 his Alma Mater, Rose Polytechnic Institute, conferred on him the honorary degree of Doctor of Engineering. He was also a member of the *Washington Society of Engineers*, and of the *Society for the Promotion of Engineering Education*.

Dr. Hood was married in 1884 to Miss Gertrude Benight, of Terre Haute, Ind., who survives, as well as three sons, Ben, Karl and Harrison, twelve grandchildren, two great grandchildren, and a brother, Arthur Hood.

Funeral services for Dr. Hood were held at his residence Saturday afternoon, April 24, with the Rev. J. Russell Clinchy officiating, and interment took place in Rock Creek Cemetery.

The Officers and Council of the Society sincerely regret the passing of so distinguished a member and express their sympathy to the members of his family who survive.

R. T. Coe Dies

The Society records with sadness the death of Ralph T. Coe, which occurred April 27, 1937, following a heart attack at his office. For over 35 years he had been active in all phases of air handling and for the past 20 years he operated The R. T. Coe Companies in Rochester, N. Y., with branches in Buffalo and Syracuse, specializing in heating, ventilating, drying and mechanical draft.

Mr. Coe was born November 29, 1882 at Medina, N. Y., and he received his early education in the public schools of Buffalo. He was associated with Buffalo Forge Co. from 1895 to 1904 and in various engineering and sales capacities with American Blower Co., Detroit, Mich., from 1905 to 1914. Mr. Coe organized Canadian Sirocco Co., Ltd., Windsor, Ont. Can., and served as director and manager during 1914-1917. He was Western New York District Manager for American Blower Co., Warren Webster &

R. T. Coe

Co., The Ric-wiL Co., Kewanee Boiler Co., Patterson-Kelley Co., and Clarage Fan Co. since 1931, and also represented Boylston Steam Specialty Co.

Mr. Coe joined the American Society of Heating and Ventilating Engineers in 1917 and had served as president of the Western New York Chapter 1919-1920 and as a member of the Board of Governors. During the World War he was a member of the Society's committee cooperating with the U. S. Food Administration, U. S. Department of Agriculture, on the drying of fruits and vegetables. Mr. Coe was very active in Rotary work and was president of the Rochester Rotary Club during 1926-1927. He was a member of many national and local associations and clubs, and was a Mason.

Surviving Mr. Coe are his widow, Ruth G. Coe, two daughters, Faye Elizabeth Green, and Ada Bethine Coe, two brothers, Ivan B. of Rochester, and C. LeRoy Coe of Detroit, and two grandchildren, to whom the officers and members of the Council express their sincere sympathy.

Funeral services for Mr. Coe were held at his home, 235 Chili Ave., Rochester, April 29.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membereship are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 38 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by June 15, 1937, these candidates will be ballotted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
BADGETT, W. H., Research Asst., A. & M. College of Texas, College Station, Texas. (*Advancement*)	F. E. Giesecke E. G. Smith	W. E. Long C. E. Griesser
BOYD, S. W., Consulting Engr., Newcomb & Boyd, Atlanta, Ga. (*Reinstatement*)	I. B. Kagey C. L. Templin	E. R. Foss K. E. Gouedy
BROCKINTON, C. E., Sales Engr., Advanced Refrigeration, Inc., Atlanta, Ga.	H. K. McCain R. F. Hahn	E. R. Foss T. T. Tucker
CARLSON, C. O., Prop., C. O. Carlson Htg. Co, Minneapolis, Minn.	H. M. Betts A. B. Algren	C. E. Lund F. B. Rowley
COOK, G. E., Vice Pres., Airconditioning, Inc., St. Paul, Minn.	H. M. Betts G. C. Morgan	Carl Gausman E. F. Jones
DONNELLY, M. A., Sales, McArdle & Co., Philadelphia, Pa.	J. H. Hucker H. L. Galson (*A.S.M.E.*)	W. J. Barber (*A.S.M.E.*) J. S. Morehouse (*A.S.M.E.*)
DOWLER, E. A., Sales Engr., B. F. Sturtevant Co. of Can., Ltd., Toronto, Ont., Can.	H. J. Church O. L. Maddux	Thomas McDonald M. W. Shears
FISHER, G. B., Vice Pres., Midwest Air Cond. Corp., St. Louis, Mo.	Paul Sodemann George Myers	Elliott Harrington D. W. McLenegan
GAUSE, H. C., Power Sales Engr., Ala. Power Co., Birmingham, Ala.	H. V. Fried C. P. Lichty	O. E. Gammill, Jr. C. L. Templin
GILBERT, L. S., Owner, Gilbert Engrg. Co., Dallas, Texas.	H. W. Skinner R. K. Werner	R. F. Taylor C. L. Kribs, Jr.
HALLAR, E. V., Mgr., Coglizer Insulation Co., Kansas City, Mo.	L. R. Chase David Caleb	E. K. Campbell L. A. Stephenson
HAUER, Fred, Pres., Fred Hauer & Co., Peoria, Ill.	H. A. Soper R. P. Whitmer	S. A. Baird J. G. Farnsworth
HEISTERKAMP, H. W., Design Engr., Bryant Heater Co., Cleveland, Ohio.	W. E. Stark L. C. Harvey	G. L. Tuve B. P. Fonda (*Non-Member*)
HERSH, F. C., Engr., Penn. Power & Light Co., Allentown, Pa. (*Reinstatement*)	I. I. Bock O. H. Mehl	G. H. Meffert J. T. West, Jr. (*Non-Member*)
JESSUP, B. H., Htg. & Vtg. Contractor, Richards & Jessup Co., Inc., Stamford, Conn.	S. A. S. Patorno A. O. Blackman	A. J. Offner M. C. Giannini
JOHNSTON, R. M., Instructor Dept. Mech. Engrg., Va. Polytechnic Inst., Blacksburg, Va.	E. B. Norris (*A.S.M.E.*) J. I. Clower (*S.A.E.*)	F. S. Roop (*A.S.M.E.*) J. B. Jones (*A.S.M.E.*)
JONES, A. T., Mech. Engr., S. A. Armstrong Ltd., Toronto, Ont., Can. (*Advancement*)	G. A. Playfair Thomas McDonald	M. W. Shears H. R. Roth
KNOWLES, E. L., Mgr., Marshall Htg. Co., Minneapolis, Minn.	H. M. Betts F. B. Rowley	C. E. Lund A. B. Algren
LABOV, MILTON, Vice Pres., B. E. Labov & Son, Atlantic City, N. J. (*Reinstatement*)	S. B. Strouse F. W. Plumber (*Non-Member*)	W. D. Howard (*Non-Member*) H. W. Ralston, Jr. (*Non-Member*)
LANDES, B. D., Mgr. Engrg. Serv. Dept., A. M. Byers Co., Pittsburgh, Pa.	F. N. Speller James Aston (*Non-Member*)	D. W. Stembil (*A.S.T.M.*) E. B. Story (*A.S.T.M.*)
LOWER, H. C., Sales Engrg. Contact, J. J. Gibbons, Ltd., Toronto, Ont., Can.	Thomas McDonald O. L. Maddux	J. W. O'Neill G. A. Playfair
MADDEN, J. J., Plbg. & Htg., The Madden Co., Roxbury, Mass.	W. E. Barnes W. T. Jones	N. J. H. Shaw E. Q. Cole (*Non-Member*)
MCCULLOUGH, J. L., Asst. Br. Mgr., National Radiator Corp., Pittsburgh, Pa. (*Reinstatement*)	D. W. Loucks J. F. Collins, Jr.	T. F. Rockwell P. A. Edwards

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
PETERSEN, C. P., Prop., Petersen Sheet Metal Works, Minneapolis, Minn.	H. M. Betts C. E. Lund	F. B. Rowley A. B. Algren
PRICE, C. F., Engr. & Draftsman, The Knapp Supply Co., Muncie, Ind.	J. E. Maynard C. E. Price	D. B. Poling F. P. Keeney
PRITCHARD, W. J., Br. Mgr., Carrier Corp., Newark, N. J.	R. E. Cherne W. A. Grant	V. S. Day R. V. D. Dunne (*A.S.R.E.*)
REINKE, L. F., Owner, Reinke Sheet Metal Works, Milwaukee, Wis.	R. G. Koch Ernest Szekely	J. H. Volk C. H. Randolph
RICHFIELD, N. H., Field Engr., Delco-Frigidaire Cond. Div., New York, N. Y.	D. J. Edwards C. E. Lewis	Albert Buenger W. H. Hutchins
SALINGER, R. J., Engr., General Refrigeration Sales Co., Kansas City, Mo.	Emil Haas, Jr. D. M. Allen	M. A. Disney T. H. Anspacher
SHERMAN, W. P., Commercial Div. Engr., York Ice Mach. Co., Decatur, Ga.	W. H. Moler L. J. Drum, Jr. (*A.S.M.E.*)	C. H. McLeod (*Non-Member*) W. M. Garrard (*Non-Member*)
SMITH, W. D., Pres., Bryant-Smith, Inc., Cleveland, Ohio. (*Advancement*)	W. E. Stark F. A. Kitchen	R. A. Wilson J. E. Kinner (*Non-Member*)
STEWART, J. P., Engr., Carrier Corp., Newark, N. J.	R. E. Cherne L. B. Mann	R. V. D. Dunne (*A.S.R.E.*) W. B. Rorison (*A.S.R.E*)
THEOBALD, ART, Research Engr., Payne Furnace & Supply Co., Beverly Hills, Calif.	J. F. Park E. H. Kendall	O. W. Ott H. M. Hendrickson
WADE, R. H., Sales Engr., Spencer Heater Co., New York, N. Y.	Jacob Miller Philip Wolf	Jacob Kessler Alfred Hering
WALLACE, G. N., Owner, George N. Wallace Co., New York, N. Y.	J. E. Axeman E. N. McDonnell	W. A. Danielson T. H. Urdahl
WIDDOWFIELD, A. S., Sales Engr., The Mercoid Corp., Chicago, Ill.	R. P. Whitmer John Jex, Jr.	L. P. Hynes W. K. Stauffer (*Non-Member*)
WILKES, G. B., Prof. of Heat Engrg., Mass. Inst. of Tech., Cambridge, Mass.	James Holt C. M. F. Peterson	F. H. Norton (*Non-Member*) W. H. Jones (*Non-Member*)
ZUROW, WILLIAM, Sales, St. Joseph Ry. Lt. H. & P. Co., St. Joseph, Mo.	T. H. Anspacher A. J. Harton	L. R. Chase D. M. Allen

Candidates Elected

In past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

COVER, E. B., Sales Engr., York Ice Mach. Corp., East St. Louis, Ill.

FIFE, G. D., Air Cond. Engr. for Archt. of the Capitol, Washington, D. C. (*Advancement*)

GUNNELL, G. T., Chief Engr., Sunbeam Htg. & Air Cond. Co., Atlanta, Ga.

HANLEIN, J. H., Asst. Secy. & Treas., Weller Corp., Washington, D. C.

HILDER, F. L., Chief Engr., Electric Furnace-Man, Inc., New York, N. Y.

KELLEY, R. D., Pres.-Treas., Sunbeam Htg. & Air Cond. Co., Atlanta, Ga.

KENDALL, E. H., Sales Engr., English & Lauer, Inc., Los Angeles, Calif. (*Reinstatement and Advancement*).

MAHON, F. B., Air Cond. & Vtg. Specialist, Duquesne Light Co., Pittsburgh, Pa.

MARTENS, E. D., Mgr. Mech. Dept., Thompson Starrett Co., New York, N. Y.

MCGEORGE, R. H., Mgr. Htg. Dept., McCord Radiator & Mfg. Co., Detroit, Mich. (*Reinstatement*)

RUMMEL, A. J., Industrial Engr., San Antonio Public Service Co., San Antonio, Texas.

SHEPARD, J. DEB., Air Cond. Repr., Consolidated Gas Elec. Light & Power Co. of Baltimore, Baltimore, Md.

SHERWOOD, L. T., Glass Technologist, Penn. Wire Glass Co., Dunbar, Pa.

SIMISON, A. L., Research Engr., Owens-Illinois Glass Co., Newark, Ohio.

SOETERS, MATTHEW, Consulting Engr., Smith, Hinchman & Grylls, Detroit, Mich.

WERNER, J. G., L. J. Mueller Furnace Co., Philadelphia, Pa.

WYLD, R. G., Executive Engr., Airtemp, Inc., Dayton, Ohio.

ASSOCIATES

CHARTERS, W. A., Htg. Sales, Can. Foundries & Forgings, Ltd., Brockville, Ont., Can.

DULL, E. J., Engr.-Contractor, Baltimore, Md.

FELDERMANN, WILLIAM, Mgr. Air Cond. Div., American Gas Accumulator, Elizabeth, N. J.

GERHARD, D. H., Power Sales Engr., Consumers Power Co., Jackson, Mich.

HINES, G. M., Chief Engr., A. & M. College of Texas, College Station, Texas.

HUGHES, F. S., JR., Estimator of Special Mill Work, Hardin Sash & Door Co., Ft. Worth, Texas. (*Reinstatement*)

LEWIS, H. F., Vice Pres., Harvey A. Dwight Oil Heat & Supply Co., Inc., Albany, N. Y.

MANDELL, T. P., Sales, Carrier Corp., Boston, Mass.

MCCAFFERTY, J. E., Dist. Engr., Petroleum Heat & Power Co., Boston, Mass.

MOFFAT, O. G., Sales Engr., Livingston Stoker Co., Ltd., Hamilton, Ont., Can.

ROY, LEO, Sales Engr., The S. W. P. Co., Montreal, Que., Can.

JUNIORS

CADY, E. F., Sales Engr., American Blower Corp., Buffalo, N. Y.

CARLSON, C. V., JR., Engr., Sidles Co., Airtemp Div., Lincoln, Nebr.

DRIEMEYER, R. C., Asst. to Chief Engr., Airtherm Mfg. Co., St. Louis, Mo.

HART, F. D., Grad Student, Cornell University, Ithaca, N. Y.

MCKINNEY, C. A., Design Engr., United Gas System, Houston, Texas.

STUDENTS

SCHOLL, H. O., Student, University of Illinois, Urbana, Ill.

News!

Controlled-Cost Air Conditioning

wins enthusiastic acceptance

SPECIAL RESIDENTIAL BULLETIN

More and more evidence is piling up daily, pointing to forced warm air heating combined with winter air conditioning as the popular heating system for new homes.

The new Delco Conditionairs, in sizes to fit any house, meet this demand perfectly. *They air condition as they heat!* They use either gas or oil. They cost no more than automatic heat alone.

And cooling equipment for Summer Air Conditioning can be added either at the time of installation—or later, provided ducts are properly designed.

THE enthusiastic acceptance given Frigidaire Controlled-Cost Air Conditioning by engineers everywhere is due not only to the technical excellence of the equipment itself but to the engineering soundness of the idea behind it.

Practically no two air conditioning jobs are exactly alike . . . either in the amount of cooling action or in the kinds and degrees of conditioning required.

Yet with Frigidaire Controlled-Cost Air Conditioning, you can specify equipment that will give the exact amount of cooling any job should have. You can specify any degree or kind of conditioning . . . from simple cooling to the most precise control of all atmospheric conditions.

And, perhaps most important of all, you can specify equipment to fit any and all types of construction.

You should have for your files complete details about this "Product of General Motors". Write to Delco-Frigidaire Conditioning Division, General Motors Sales Corp., Dayton, Ohio.

What Controlled-Cost Air Conditioning Means to Engineers

It allows you to specify . . .

1. A system that gives the desired atmospheric conditions—your client pays *only* for what he needs.
2. Equipment of exactly the right size and capacity for your client—neither too small, which would mean unsatisfactory service; nor too large, which would be wasteful and costly.
3. A *method* of installation that suits any building—whether remodeled or new, owned or rented, therefore controlling the ultimate cost.
4. *More* cooling action with *less* current consumption. Hence a control over operating costs.
5. Dependable, *proven* equipment for low maintenance cost.

And gives you and your client a presentation of *all* the facts, so that you will know *and can therefore control* the entire cost.

It Pays to Talk to **DELCO-FRIGIDAIRE**

The Air Conditioning Division of General Motors

AUTOMATIC COOLING, HEATING AND CONDITIONING OF AIR

Save Money on Your Piping and Gain These Advantages with WELDOLETS·THREDOLETS

Eliminate all preliminary layouts of main pipe and branch. No templets are necessary. All cutting, threading, forming and fitting of main pipe eliminated. Permit making any type of outlet—tee, side-outlet, cross, etc. All joints leak-proof and full pipe strength. Heavy external rib compensates for loss of strength in main pipe when hole is cut. Funnel-shaped outlets of WeldOlets and ThredOlets eliminate "dead spots" providing free flow conditions. Turbulence and friction reduced to a minimum. Make neater, more workmanlike joints. Adapted to new construction or for maintenance. Suitable for use in high pressure lines or where high temperatures are used. Available with outlet sizes of ½" to 12" for all standard pipe sizes. Supplied in Wrought Iron, Toncan Iron, Brass, Bronze, Copper, Nickel, Monel, Everdur etc. to meet your specific needs. Installed by Oxy-Acetylene or Electric-Arc welding.

"WeldOlets" and "ThredOlets" are fully covered by U. S. and foreign patents and are trade marks registered in the United States Patent Office.

Write for Bulletin WT21 giving full details.

Bonney Forge and Tool Works

Cedar & Meadow Sts., Allentown, Pa.

Stocked by Leading Distributors

WELDOLETS·THREDOLETS

Conventions and Expositions

American Water Works Association: Annual convention, June 7-11, Hotel Statler, Buffalo, N. Y.

National Warm Air Heating and Air Conditioning Association: Mid-year convention, June 9-10, Hotel Cleveland, Cleveland, Ohio. Managing Director, Allen W. Williams, 50 W. Broad St., Columbus, Ohio.

National Association of Building Owners and Managers: Convention and exhibit, June 21-24, Buffalo, N. Y. Secretary, Robt. B. Beach, 134 S. LaSalle St., Chicago, Ill.

American Society of Agricultural Engineers: Annual meeting, June 21-24, University of Illinois, Urbana, Ill. Headquarters Office, St. Joseph, Mich. (Subjects to be discussed include artificial curing of forage crops, use of insulation under high humidity conditions; heating, ventilating and humidity control in tobacco-curing barns; and the drying of hay and other crops.

American Society of Heating and Ventilating Engineers: Semi-annual meeting, June 24-26, New Ocean Hotel, Swampscott, Mass. Secretary, A. V. Hutchinson, 51 Madison Ave., New York, N. Y.

American Society for Testing Materials: Annual meeting and exhibit of testing apparatus, June 28-July 2, The Waldorf-Astoria, New York, N. Y. Headquarters Office, 260 S. Broad St., Philadelphia, Pa.

Universal Craftsmen Council of Engineers of the World: Power show and mechanical exposition, August 3-7, Hotel Stevens, Chicago, Ill. General Chairman, W. T. Walters, 7965 Phillips Ave., Chicago, Ill.

National Association of Power Engineers: Convention, August 30-September 3, Jefferson Hotel, St. Louis, Mo.

American Gas Association: Annual convention, September 27-October 1, Hotel Cleveland, Cleveland, Ohio. Headquarters Office, 420 Lexington Ave., New York, N. Y.

Association of Iron and Steel Engineers: Convention and exposition, September 28-October 1, Hotel Stevens, Chicago, Ill. Headquarters Office, Empire Bldg., Pittsburgh, Pa.

Chicago Exposition of Power and Mechanical Engineering: October 4-9, International Amphitheatre, Chicago, Ill. Under management of the International Exposition Co., Grand Central Palace, New York, N. Y., manager of the biennial National Expositions of Power and Mechanical Engineering, the next of which will be held in New York City December 5-10, 1938.

American Welding Society: Annual convention, week of October 18, Hotel Traymore, Atlantic City, N. J. Headquarters office, 33 W. 39th St., New York, N. Y.

Chemical Industries Exposition: December 6-11, Grand Central Palace, New York, N. Y.

Fifth International Heating and Ventilating Exposition: January 24-28, 1938, Grand Central Palace, New York, N. Y. Manager, Charles F. Roth, International Exposition Co., Grand Central Palace, New York, N. Y.

Purdue Offers Summer Course in A. C.

Purdue University, Lafayette, Ind., will offer this year for the first time summer school courses in mechanical refrigeration and in air conditioning. These courses will be intensive in that they will be completed in twenty days each. The course in refrigeration will begin June 18 and end July 10, while the course in air conditioning will start July 12 and end July 31.

They are designed for teachers, engineers, salesmen, students and others interested in acquiring a working knowledge of the principles and general applications of refrigeration and air conditioning. The fee for each course is $18.00. Class work will be given every morning (except Sunday) for three hours. University credit will be given to those who successfully pass examinations. Those who do not desire credit may omit the examinations.

☆ U. S. GOVERNMENT has Non-Rust Tanks of Everdur Silicon Bronze in new Coast Guard Station at San Diego

Three 36" diameter EVERDUR tanks are now in use at the new U. S. Coast Guard Station at San Diego, Cal. They were made for Crane Co., San Diego Branch, by the Los Angeles Boiler Works to comply with U. S. Government Specification WWP-541.

U. S. Coast Guard Station, Marine Base, Lindbergh Field, San Diego, Cal.

NOTHING was spared in assuring the absolute efficiency of this splendid Coast Guard Base. For these heater shells, The American Brass Company's copper-silicon alloy, EVERDUR, assures long life...freedom from rust ...and the elimination for all time of troublesome rust-repairs and heavy replacement expense.

EVERDUR is a sturdy alloy. It combines the corrosion-resisting qualities of copper with the strength of steel. Yet its fabricating cost is relatively low, and it is readily welded by usual methods. EVERDUR storage water heaters for every purpose are obtainable from leading equipment manufacturers throughout the country.

ANACONDA

Everdur Metal

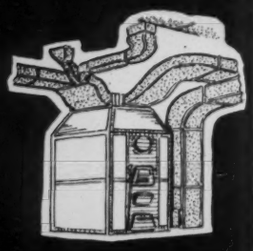
RUSTLESS AS COPPER
STRONG AS STEEL

37164

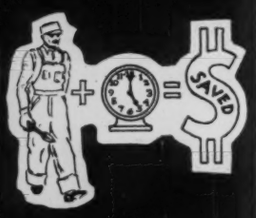
THE AMERICAN BRASS COMPANY, General Offices: WATERBURY, CONNECTICUT
Offices and Agencies in Principal Cities • In Canada: ANACONDA AMERICAN BRASS LTD., New Toronto, Ont.

*EVERDUR is a trade-mark of The American Brass Company, registered in the United States Patent Office.

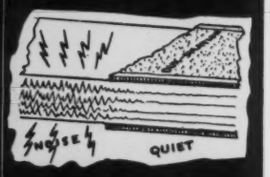
PERFECT INSULATION



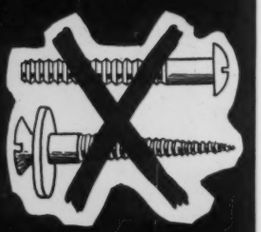
ABSOLUTE FLEXIBILITY



LABOR SAVING



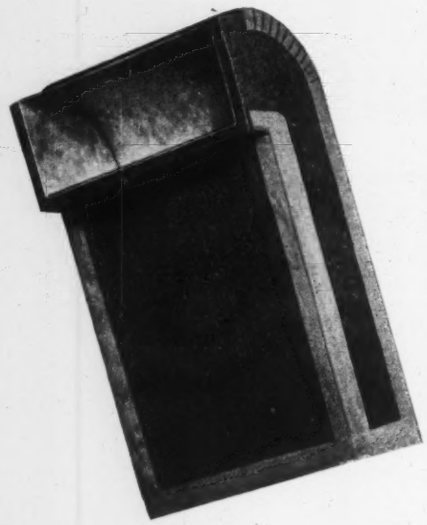
SOUND ABSORPTION



NO SCREWS, LUGS, BOLTS OR WIRE

UNIT PACKAGE — NOTHING ELSE TO BUY

AROUND ANY BEND
with
DUX-SULATION
FLEXIBILITY



THE Flexibility of Dux-Sulation is one of its outstanding features, for it cuts labor costs to a minimum. It can be applied in larger strips without the use of lugs, screws, bolts or wire. A special adhesive and a finishing tape hold it securely in place for permanent and efficient insulation service. Dux-Sulation is a specially constructed non-rotting, non-deteriorating, non-chipping or crumbling Fibre Felt with a hard reinforced outer surface. It is moisture proof, lasts indefinitely, and its clean, neat and unmarred finish will add materially to the attractiveness of any duct job. Dux-Sulation comes 36 inches wide and 3/8 inch thick. It is put up in convenient-to-handle rolls of 100 square feet. The Dux-Sul-Glue and the Dux-Sul-Tape shipped with it is all you need to install Dux-Sulation. No "extras to buy." Dux-Sulation is the universally accepted all purpose duct insulation. It will save many dollars in installation cost, original cost, and in operation. Send for samples and complete information at once.

Dux-Sulation A PRODUCT OF
GRANT WILSON INC
Taylor and Karlov Sts. Chicago, Illinois

Booklets, Reports and Papers

Engineering Report on Air Conditioning Railroad Passenger Cars Issued

The equipment research division of the *Association of American Railroads* has just issued the second of its series of reports on air conditioning railroad passenger cars. The summary report, published last fall, dealt with those factors conceived to be of direct concern to the railroad executive; the present engineering report is designed to meet the needs of the air conditioning engineer and railroad operating personnel, and in it are presented the results of laboratory and road performance tests of air conditioning systems, drives, and cars.

The extensive investigation upon which these reports have been based comprises surveying the performance of equipment in service; determining the cooling capacity and power consumption of air conditioning systems; determining the mechanical efficiency of various drive mechanisms; studying air requirements, treatment, diffusion, and related matters; determining the cost of air conditioning per 1000 car-miles; and a study of the factors relating to passenger comfort. The present report is divided into four parts—tests of air conditioning systems, tests of drive mechanisms, hot room tests of air conditioned cars, and road performance tests. Detailed data are presented, and many charts and graphs are included.

Findings Summarized

Major findings of the research program are briefed in an introductory section of the report.

In this summary are included data on installation, operating, and maintenance costs, and on fixed charges, for the ice activated, electro-mechanical compression, steam ejector, and direct mechanical compression systems, as well as total costs per 1000 car-miles, including fixed charges, operating, and maintenance costs for cooling seasons of different lengths, an average train speed of 50 mph, and a total mileage of 150,000 car-miles per year. For a three month cooling season, this total cost figure is given as $9.48 for the ice activated system; $12.63 for the electro-mechanical compression system; $14.25 for the steam ejector system; and $14.49 for the direct mechanical compression systems. The corresponding figures for a 10 month cooling season are $13.14 for the electro-mechanical system; $14.62 for the steam ejector; $14.91 for the direct mechanical; and $16.95 for the ice activated system.

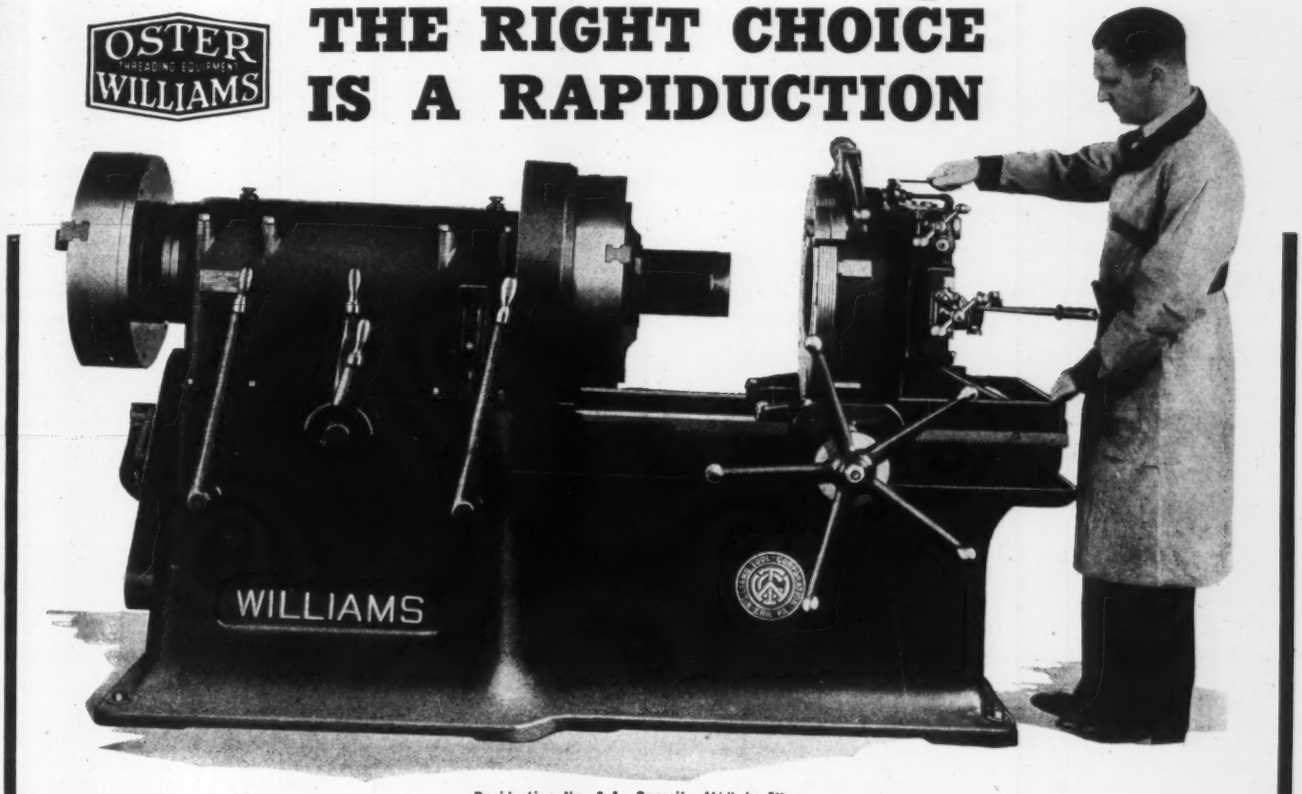
The methods of air delivery, in descending order of their effectiveness, rank as follows, according to the report: Inside center duct; double outside duct; single outside duct; center bulkhead; and double end bulkhead. As to insulation, it is pointed out that the refrigeration capacity required to absorb the heat transferred through the car structure in approximately still air with a 20 deg difference between inside and outside temperatures was 1.3 tons for the best insulated car and 3.1 tons for the worst insulated.

Desirable Conditions

Probable desirable conditions in air conditioned cars should be, the report says, an inside temperature of from 72 to 76 deg, with a minimum of 70 and a maximum of 80; inside relative humidity from 40 to 60 per cent, with a minimum of 30 and a maximum of 70; a difference in average temperatures not in excess of 2 deg between any two positions in the car, with the average temperature at ankle level not more than 1 deg below that at head level of the person seated; air movement of from 25 to 75 fpm; and a minimum of 10 cfm of fresh air delivered through the cooling system per passenger for normal conditions, and 12 to 15 cfm where smoking is permitted. Full automatic temperature and humidity control is essential, with modulating

OSTER THREADING EQUIPMENT WILLIAMS

THE RIGHT CHOICE IS A RAPIDUCTION



Rapiduction No. 6-A, Capacity 1½" to 6".

● In the production shop where time means money and accuracy is a "MUST," Rapiduction Pipe-Threading and Cutting-off Machines are prime favorites with men and managements.
No job can balk a Rapiduction.

Its dual clutch control and rear gripping chuck make it practical for use in making up fittings and flanges. One die head handles all sizes and one set of dies threads all sizes of same pitch. Write today for special bulletin.

THE OSTER MFG. CO.

SALES OFFICE:
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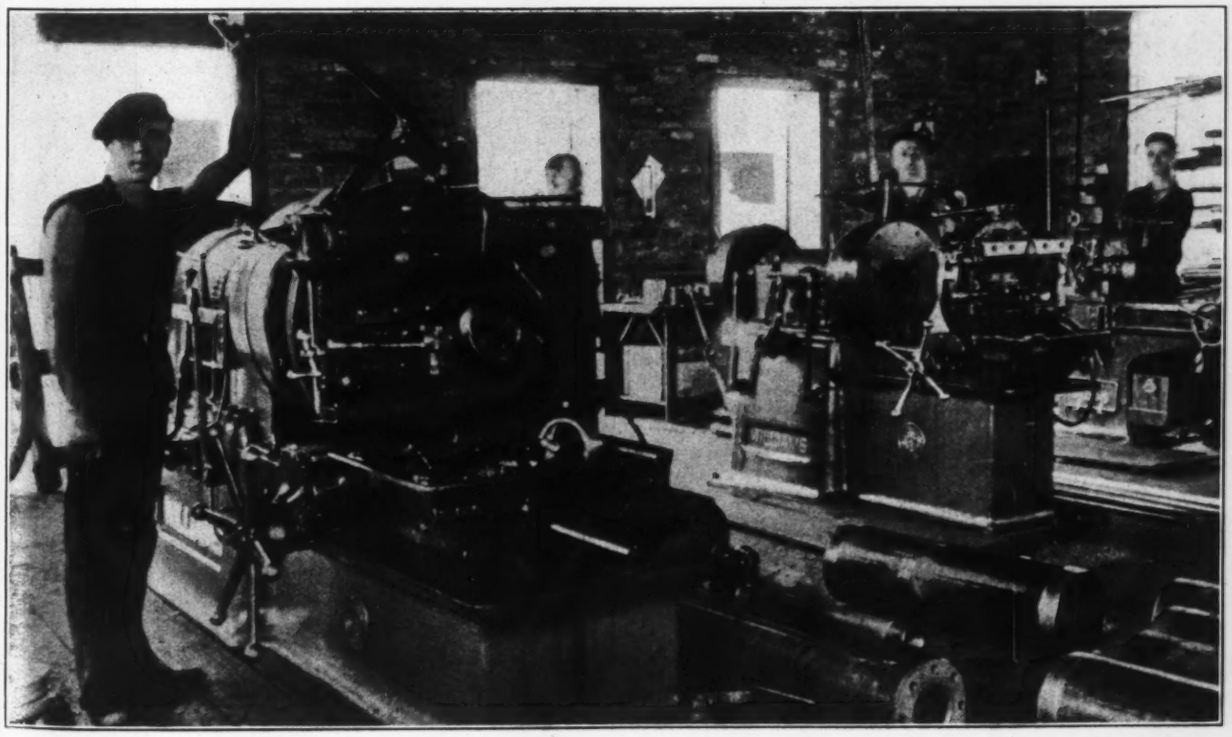
FACTORIES:
Erie, Penna., and Cleveland, Ohio
New York City Showroom and Office
292 Lafayette St.

Threading Headquarters Since 1893

A jobbing pipe-shop with three Rapiductions, Nos. 2, 6 and 12, on the job.



control desirable. The control system should coordinate heating and cooling.

A number of phases of the subject are recommended for further study. Cost of batteries and their servicing is a major item. Many different kinds of lubricating oils and greases are being used, and perhaps some standardization is advisable. Careful consideration should be given to standardization of terminal equipment and practices. Insulation is so important studies should be made of the most suitable kind, economic limits of the amount to use, and whether reductions in heat transmission can be made by changes in car construction. An exhaustive and thorough study of filters should be made.

As 15.4 per cent of unfavorable comments by passengers referred to heat from the sun through windows, a study of this subject should be made. Extensive studies are required before the odor problem can be adequately solved.

This report, paper bound, comprises over three hundred 8½ by 11 in. pages, and is available from the *Association of American Railroads*, 59 E. Van Buren St., Chicago, Ill., for $1.50 per copy to member roads and for $3.00 to others.

Briefly Noted

Principles of Economical Heating—A new edition of this well-known and useful booklet on economical heating of buildings published by the *National District Heating Association*, 1317 Spruce St., Philadelphia, Pa., in collaboration with the *National Association of Building Owners and Managers*. Prepared by the *N. D. H. A.* commercial relations committee, of which Leonard S. Phillips is chairman. 60 pp., 6x9 in. Price, 25c.

Silicosis and Allied Disorders—History and Industrial Importance. Bulletin No. 1 in the medical series of the *Air Hygiene Foundation*, 400 Fifth Ave., Pittsburgh, Pa. Prepared for the Foundation's medical committee, of which Dr. A. J. Lanza is chairman. This bulletin is the result of a comprehensive review of the literature and an evaluation of the knowledge to date and concludes with a summary of what is known about pneumoconiosis (especially silicosis), what is controversial, and in what particular directions research should be supported. 178 pp., 6x9 in.

Air Conditioning Code—An air conditioning code recommended for use in the State of Wisconsin, prepared by a committee of which E. Szekely, 1817 S. 66th St., Milwaukee, Wis., is chairman. Includes information on design load factors, design conditions, ventilation requirements, air circulation and distribution, duct capacity, apparatus rating, requirement for selection of refrigerating equipment, refrigerants, noise control, guarantees and exceptions. 8 pp., 8½x11 in. Price, 25c.

Exhaust Systems—Fundamentals relating to the design and operation of exhaust systems (preliminary edition). Published by the *American Standards Association*, 29 W. 39th St., New York, N. Y. 27 pp., 7¾x10½ in. Price, 40c.

Sheet Copper Handbook—Second edition, revised 1937, including information for architects and sheet metal workers, to whom it is available on request on business letterhead to the publisher, *Copper and Brass Research Association*, 420 Lexington Ave., New York, N. Y.

Arbitration—April, 1937, issue of "The Arbitration Journal," published by the *American Arbitration Association*, 8 W. 40th St., New York, N. Y., featuring a symposium on arbitration in the real estate and construction industry. Price, 60c.

Properties of Pipe Tables

In the table of properties of "Schedule 10" pipe published on page 534, October, 1936, issue and in "summary sheet" form on page 247, April, 1937 issue, the area of metal in sq in. for the 24 in. O. D. size should read 18.65. The internal cross sectional area for the 20 in. O. D. size should read 298.65; for 24 in., 433.74; and for 30 in., 677.76. The respective sq ft figures should read 2.074, 3.012, and 4.707. Also, the respective figures for weight of water per ft of pipe should read 129.42, 187.95 and 293.72 lb.

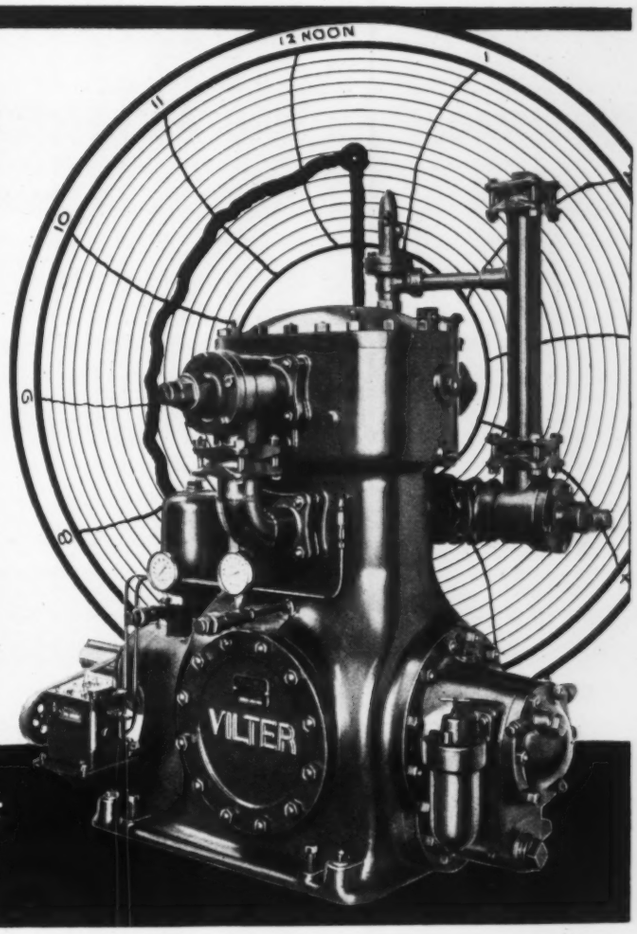
AMAZING HORSEPOWER *Economy* with a VILTER "FREON 12" COMPRESSOR!

Investment and subsequent cost of operation are vital factors in the prudent choice of any type of equipment.

By the elimination of many factors which require power, and by the refinement of others, the "VILTER FREON 12 COMPRESSOR" has shown amazing ability to deliver oversize performance at extremely low power cost.

A new and revolutionary shaft seal, plus other outstanding features contributing to efficient and economical performance with modern low-pressure refrigerants, make this "Freon 12" stand head and shoulders above the field in performance, dependability and economy.

It is another engineering achievement typical of VILTER. Let us tell you more about it.

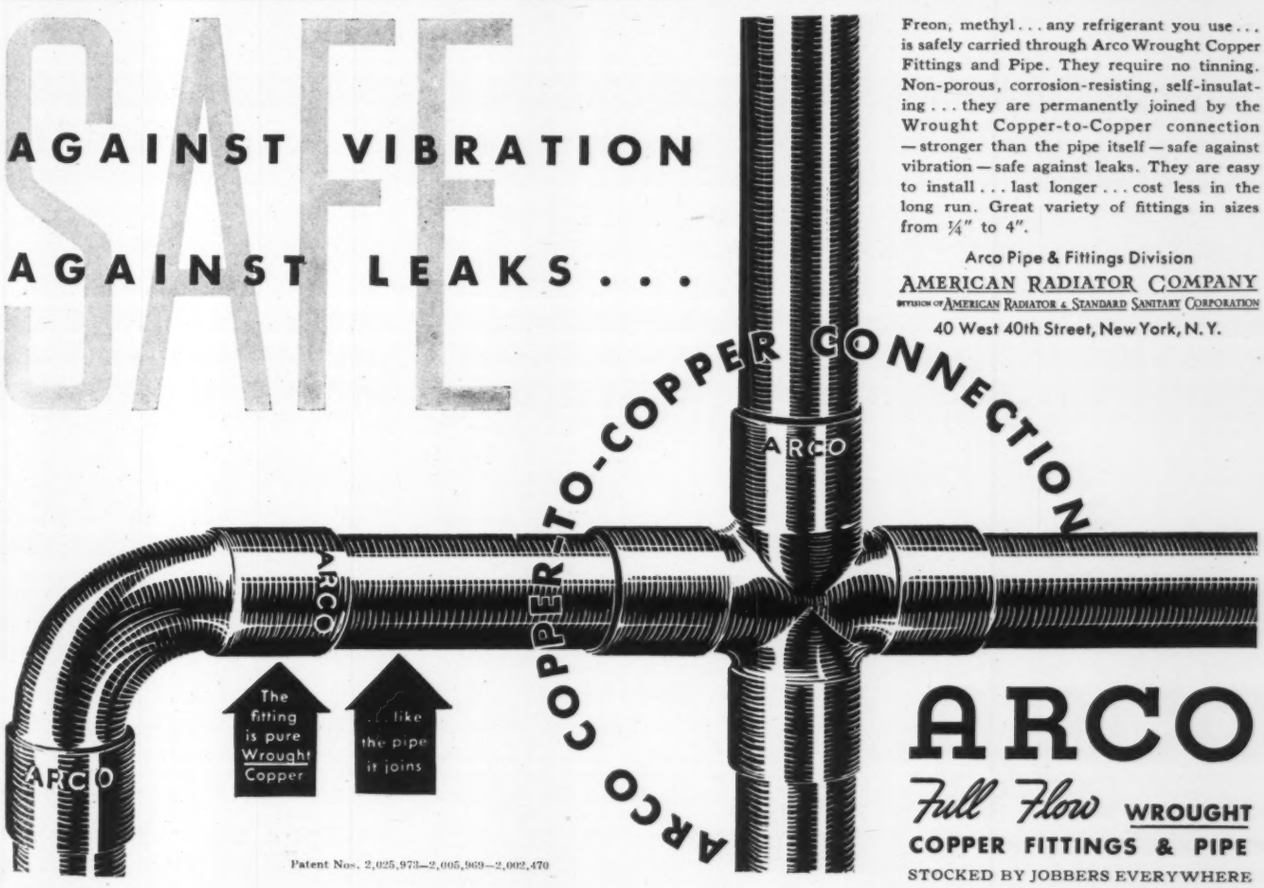


THE VILTER "SINCE 1867" MANUFACTURING COMPANY
2148 South First Street
MILWAUKEE, WISCONSIN
REPRESENTATIVES IN PRINCIPAL CITIES

SAFE AGAINST VIBRATION AGAINST LEAKS . . .

Freon, methyl . . . any refrigerant you use . . . is safely carried through Arco Wrought Copper Fittings and Pipe. They require no tinning. Non-porous, corrosion-resisting, self-insulating . . . they are permanently joined by the Wrought Copper-to-Copper connection — stronger than the pipe itself — safe against vibration — safe against leaks. They are easy to install . . . last longer . . . cost less in the long run. Great variety of fittings in sizes from ¼" to 4".

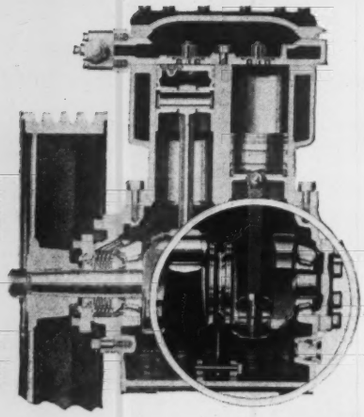
Arco Pipe & Fittings Division
AMERICAN RADIATOR COMPANY
DIVISION OF AMERICAN RADIATOR & STANDARD SANITARY CORPORATION
40 West 40th Street, New York, N. Y.



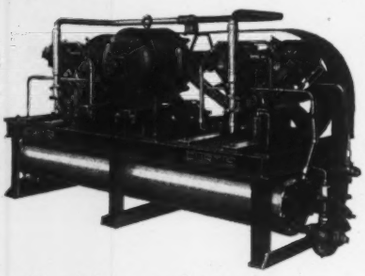
Patent Nos. 2,025,973—2,005,969—2,002,470

ARCO
Full Flow WROUGHT
COPPER FITTINGS & PIPE
STOCKED BY JOBBERS EVERYWHERE

CURTIS
Condensing Units



"CENTRO-RING" LUBRICATION— —only CURTIS COMPRESSORS have it



THERE'S just one moving part in Curtis "Centro-Ring" lubrication—a simple, free running ring that picks up oil from the crankcase and distributes it by centrifugal force under positive, uniform pressure to all points of lubrication —no gears — no plungers — nothing to get out of order or require service.

Engineering superiorities of this kind have built the Curtis reputation for reliability, long operating life and low cost of upkeep.

The Curtis line contains 41 air-cooled units, from 1/6 HP to 5 HP, inc., and 45 water-cooled units from 1/3 HP to 30 HP — a size and type to balance most any installation you might make. Curtis has built condensing units for 15 years and has made engineering equipment for 82 years — a record of success that in itself is sufficient warranty of the high-quality of Curtis units of today.

CURTIS Engineering Refinements

- Centro-Ring — one piece, positive pressure lubrication
- Timken Tapered Roller Bearings
- V-type radial compressor
- Water-jacketed compressor heads and cylinders
- Built-in oil separator with automatic return
- Drop forged, heat treated crankshafts and rods
- Balanced syphon, bellows seal
- Automatic water valves

Represented in Canada by
Canadian Curtis Refrigeration Co., Ltd.
20 George St., Hamilton, Ont., Can.

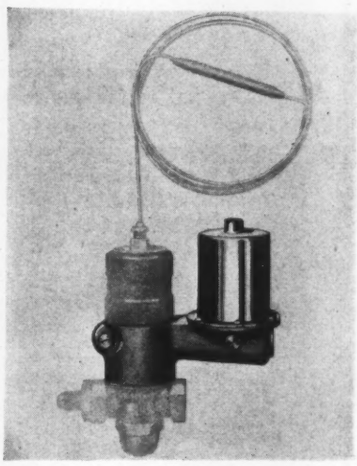
CURTIS REFRIGERATING MACHINE CO.
"Builders of Condensing Units since 1922"
1950 Kienlen Ave., St. Louis, U. S. A.

Equipment Developments

For your convenience in obtaining more information about any of this equipment, see coupon on page 144. Add the new products and companies listed here to your Directory Section which you received in your January, 1937, HEATING, PIPING AND AIR CONDITIONING and thus keep your records of sources of supply up to date throughout the year . . . Single asterisk (*) indicates equipment not listed in Directory Section; double asterisk (**) equipment and manufacturer not listed.

Modulating Attachment for Thermostatic Expansion Valves

No. 1231—A modulating attachment for thermostatic expansion valves has been developed to cut the installation and maintenance cost with such valves. The modulating attachment is applied (in the manufacturer's factory) to various standard makes of thermostatic expansion valves to form a complete unit that is controlled by a distant thermostat as well as by the usual bulb on the outlet of the evaporator. The bulb operates the expansion valve in the usual way. When maximum cooling is not required, the controlling instrument sensing room or duct temperature or humidity, gradually throttles the expansion valve, regardless of the action of the bulb, to reduce the flow of refrigerant.



This arrangement is believed to be an improvement over methods involving starting and stopping the compressor direct or placing a stop valve in the liquid line and operating a valve in this line with positive action. The device provides a means for controlling the air stream temperature at the outlets of the cooling units to prevent drafts and maintain comfortable conditions.

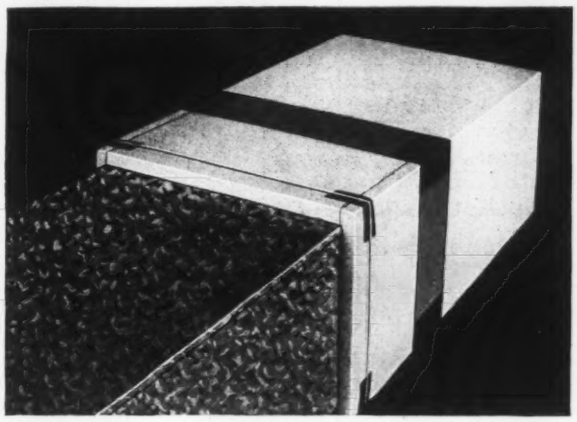
The valve is operated by a seamless metal bellows enclosed in a strong metal top and may be actuated by a pneumatic thermo stat or humidistat.—Johnson Service Co., 507 E. Michigan St., Milwaukee, Wis.

New Line of Convection Heaters

No. 1232—New "Streamaire" copper convection heaters are available in free standing and wall hung cabinet types, the latter particularly suitable for hospitals, institutions, office buildings, and stores where floors are scrubbed regularly. Fully recessed or partially recessed models are also offered, as are plastered in types.

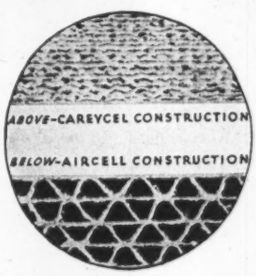
Large elliptical copper tubes are used in the heating elements, and according to the manufacturer, the size and shape of these tubes make the units suitable for use on one pipe steam systems as the condensate can flow along the bottom of the tube leaving most of the tube surface exposed to the steam. It is stated that over 95 per cent of the tube surface is in direct contact with the air because of the tube shape. The fins are stamped from copper, vertically crimped for strength, and metallically bonded to the tube. The heating element is made to withstand operating pres-

for HIGHER EFFICIENCY *in Air Conditioning*



Use Careycel

THE MODERN DUCT INSULATION FOR AIR CONDITIONING SYSTEMS



While Careycel is outstandingly more efficient, as proved by impartial tests in Mellon Institute, it sells in the same price range as air-cell. Note smallness of air spaces as compared with air-cell — a unique advantage made possible by Carey's exclusive process of "Cellizing" the asbestos sheets.

Careycel combines the low heat transmission of organic structures, such as cork and hair, with the rot-proof, fire-proof qualities of asbestos. Retards infiltration of outside warm air through pipes; reduces cold air loss through pipe walls to minimum. Reduces infiltration of outside warm air and cold air loss, through pipe walls, to minimum. Careycel shrinkage is only 1/10th that of best air cell, eliminating loss of cool air through cracks and joints, makes a smoother finished job. Available in convenient size sheets, easy to fit and apply. Write today for samples and details of Careycel's many superior qualities. Address Dept. 25.

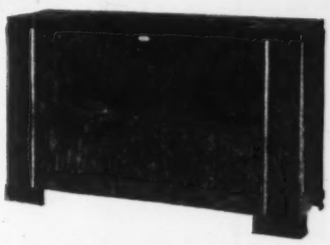
THE PHILIP CAREY COMPANY

Dependable Products Since 1873

Lockland, Cincinnati, Ohio

BRANCHES IN PRINCIPAL CITIES

Careycel ASBESTOS INSULATION



Cabinet measures 39⅜" long, 23¾" high and 13⅜" wide. Finished in rich burl walnut and velvet black enamel.

A Low Cost Air Conditioning System

That Does Everything

The High Cost Ones Do

LET'S be specific. This Burnham System does not at present include Summer cooling. It furnishes only Winter air conditioning. It is workable with any new or old radiator system of water or steam.

The Conditioners take the place of certain radiators, whether free standing or recessed. In a noiseless way, they filter-clean, humiditfy, heat and circulate the air. Equipped with both automatic and hand controls. Each unit can have its own thermostat, or be under control of a main one. Individual Unit thermostats give separate control for each room.

In Summer the Conditioner fan can be run to circulate the dead air. Automatic snap-on switch prevents fan from running in Winter, when heat in system is intermittent.

Water supply to evaporating container is also automatically controlled. Front of cabinet easily removable to reach motor. Filter screens are coarse and fine layers of spun glass, oil impregnated to catch finest dust and pollen.

Connection to system's piping is same as for a regular radiator. Any steam fitter can do it.

Simple as this Burnham Conditioner is, it does everything other Winter conditioners do, but the cost of equipment and installation is much lower.

Send for Catalog. Get full facts. See for yourself.

BURNHAM BOILER CORPORATION

Irvington, New York Zanesville, Ohio

Export Department
116 Broad Street, New York

Burnham Boiler

ADJUSTING NUT OF RIDGID PIPE WRENCH

Always Spins Easily

IN ALL SIZES 6" TO 60"

UNCONDITIONAL GUARANTEE
If this Housing ever Breaks or Distorts we will replace it Free
THE RIDGE TOOL CO. ELYRIA, O.

Guaranteed housing won't warp so the adjusting nut never binds. Wide open, it doesn't clog with sand and mud.

Add to that the saving of 75% of your wrench repairs because of the unconditionally guaranteed housing; the no-slip no-lock replaceable jaws of chrome molybdenum alloy, hook jaw with handy pipe scale—a wrench that has earned its world-wide popularity.

Ask Your Jobber

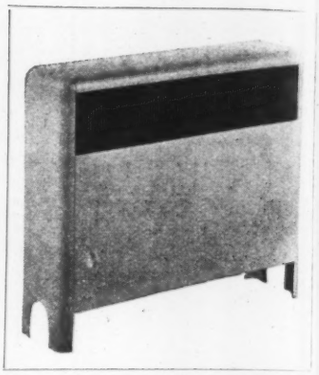
THE RIDGE TOOL CO.
ELYRIA, OHIO, U. S. A.

RIDGID PIPE TOOLS

You'll like this End pattern for pipes against flat surfaces and in tight places, plus the many advantages of the regular RIDGID

sure of 50 lb per sq. in. saturated steam or 100 lb per sq in. hydrostatic.

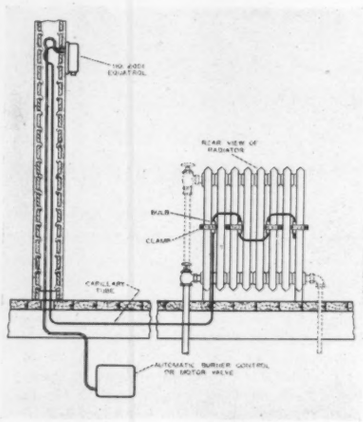
Various grille designs are available. Complete capacity data for steam or hot water, gravity and forced circulation, may be had.—Young Radiator Co., 709 Mead St., Racine, Wis.



Thermostat Responds to Room and Radiator Temperature

No. 1233—New low voltage thermostat ("Arco Equatrol") designed to eliminate stratification and "lags" in heat pick-up and at the same time insure a constant supply of radiant heat, has been developed for steam, vapor or vacuum heating systems. It responds to radiator as well as room temperature.

Four small bulbs, attached to four different sections of a radiator or convector in order to get the average radiator temperature, operate to keep the radiator always warm by never allowing the radiator iron to become cold. The room thermostat operates in the usual way, while the radiator attachment, by keeping the radiator warm, eliminates lag by calling for heat before the room temperature has dropped below the comfort point.

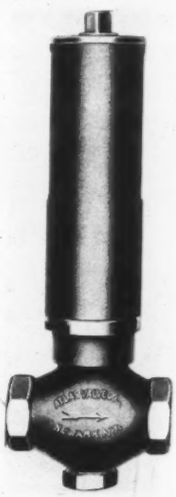


Room thermostat bulb is designed to operate the electrical contacts on a change of room temperature of 1/5 deg. or plus or minus 1/10 deg, while the radiator bulbs are designed to operate the contact points on a change of radiator temperature of 10 deg, or plus or minus 5 degrees.—American Radiator Co., 40 W. 40th St., New York, N. Y.

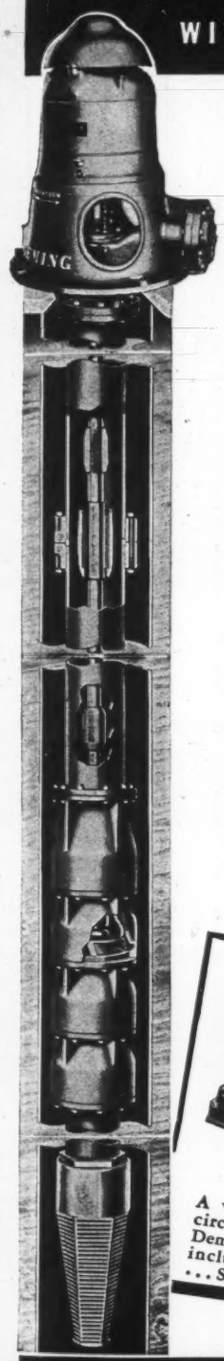
High Pressure Reducing Valve Improved

No. 1234—Improved forged steel body high pressure reducing valve for oil as well as water and air has internal metal parts entirely of stainless steel. A formed packing of special material has been adopted which is immune not only to water but to oil and other fluids commonly used in hydraulic machinery, says the maker.

The pressure on the seat is balanced by a piston, with the result that variations in high initial pressure have little effect on the reduced pressure. It is recommended that where the reduction is from a high to an extremely low pressure it be performed in two stages, using a lower pressure valve for the second stage.—Atlas Valve Co., 282 South St., Newark, N. J.



LOW COST
Well Water Air Cooling
WITH DEMING TURBINES



Where well water is available, a Deming Deep Well Turbine Pump provides an economical, highly efficient solution to the air cooling problem. Low installation cost... low operating cost... and exceptional durability insure utmost satisfaction with Deming Turbine Pumps.

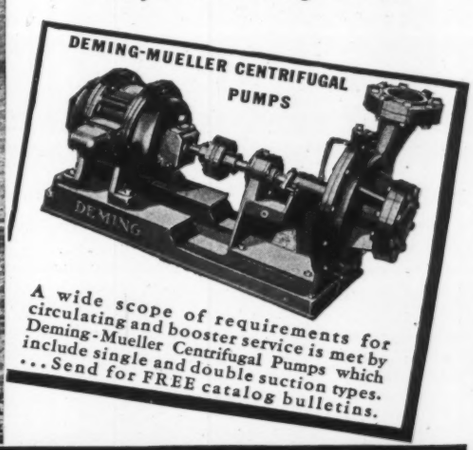
FEATURES

Water lubrication is positive and continuous as the water being pumped provides a constant lubricant for all bearings.

Adjustable impeller clearance makes it easy to sustain original high efficiencies without necessity of removing pump from well.

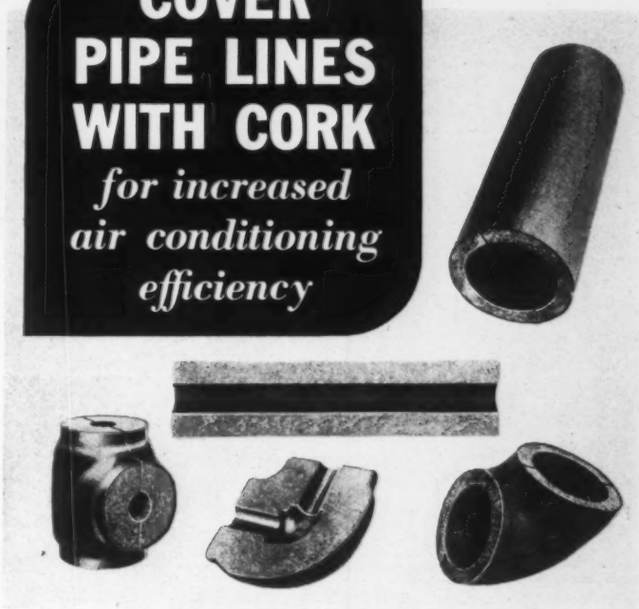
Cutless rubber bearings and *stainless steel bearing sleeves* are among many additional features of Deming Deep Well Turbine Pumps.

Send for FREE *catalog bulletin!*


DEMING-MUELLER CENTRIFUGAL PUMPS
DEMING
A wide scope of requirements for circulating and booster service is met by Deming-Mueller Centrifugal Pumps which include single and double suction types. ...Send for FREE catalog bulletins.

DEMING *Pumps*
THE DEMING CO. • SALEM, OHIO

COVER PIPE LINES WITH CORK
for increased air conditioning efficiency



NOVOID CORK COVERING is delivered ready to fit all standard pipe sizes. Photographs show 8″ pipe size cork covering ready to apply.

NOVOID Cork Covering provides long-life, low cost insulation for all types of air conditioning piping. Made of clean, dry cork granules, Novoid is machined to fit cold lines snugly, leaving no air spaces for moisture to accumulate. For added protection, it is moisture-proofed with a coating of asphalt mastic.

Novoid Cork Covering is available as sectional covering for pipes, welding ells (standard or long radius, 90 or 45 degrees), and other fittings. It is delivered ready to fit all standard pipe sizes. When you specify Novoid, you are assured of a lasting, tailor-made job that will soon repay its cost in increased operating efficiency of air conditioning units. Let us tell you more about Novoid Cork Covering. Just mail the handy coupon below or write Cork Import Corporation, 330 West 42nd Street, New York City.

BRANCH OFFICES: Chicago, 400 West Madison Street; Boston, 248 Boylston Street; Philadelphia, 1524 Chestnut Street

NOVOID CORK PIPE COVERING

FOR YOUR CONVENIENCE

CORK IMPORT CORPORATION,
330 West 42nd Street, New York City.

Please send me complete details of Novoid Cork Covering for air conditioning pipe lines.

Name .....

Street .....

City and State.....

52 BUILDING OWNERS SAVE $12,187.98 IN FEBRUARY 1937

Webster Heating Modernization Saves Average of $234 per Building in Short Month

NO "STARVING" TO GET RESULTS

Owners Report on Effectiveness of Webster Moderator and Hylo Systems

MEETS DIVERSE HEATING NEEDS

Camden, N. J.—Fifty-two buildings heated by Webster Systems of Steam Heating reported reductions in heating costs of $12,187.98 during February, 1937, an average saving of $234.19 for this short month.

These savings facts are based on heating results in 52 buildings for which figures were readily available—buildings representative of the hundreds heated by Webster Moderator and Hylo Systems of Steam Heating. Following are the actual fuel savings:

10,222....gallons of fuel oil

597.91....tons of coal

14,038,168....lbs. of steam

Economy is seldom the main objective of a Webster Heating Modernization Program. Included among the buildings reporting substantial February savings are hotels and apartment houses where perfect heating comfort for guests comes before any consideration of cost; department stores where an increase in heating expense is a wise investment if it makes the store more comfortable for customers; office buildings where heating satisfaction is an important factor in building occupancy; industrial plants, medical buildings, educational institutions, all with heating needs more important than economy.

In these 52 buildings, regardless of the heating method used—oil, coal or purchased steam—and regardless of the heating problems involved, Webster Systems provided perfectly balanced heating service during the severest weather. There is no "starving" of the systems to get results. Steam savings represent an extra advantage after the heating problem in each building has been solved.

Webster Central Controls are of the throttling type, delivering less heat in mild weather. There is no intermittent heating—no so-called "cold 70" or overheating resulting from an off-and-on control.

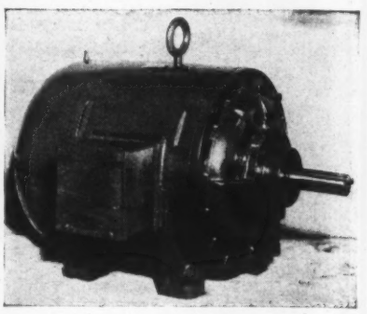
If you are interested in heating new buildings, or in improved heating service and lower heating cost in your present building address WARREN WEBSTER & CO., Camden, N. J. Pioneers of the Vacuum System of Steam Heating Representatives in 60 principal U.S. Cities—Est. 1888

Asbestos Sheet Packing Resists Oil, Refrigerants

No. 1235—Sulphur free, neoprene bonded, compressed asbestos sheet packing (known as "Prenite 20M") offers increased resistance to oil, as well as to such refrigerants as "Freon," methyl chloride and sulphur dioxide, the maker states. Absence of sulphur in its composition makes for a packing that will not corrode metal parts.

Available in 50x50 in. sheets in 1/32, 1/64 and 1/16 in. thicknesses.—Mechanical Rubber Goods Div., The B. F. Goodrich Co., 548 S. Main St., Akron, Ohio.

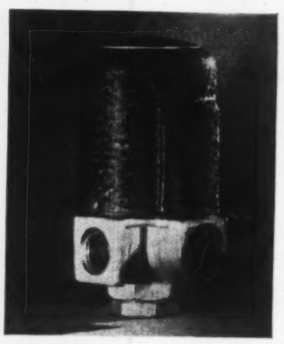
New Fan Cooled Squirrel Cage Motors



No. 1236—Protected against abrasive dust, moisture and corrosion, a new design of dual ventilated fan cooled squirrel cage motors has been announced. The motor frame contains two separate sets of air ducts, one internal and the other external.

The heat exchanger principle is used for cooling. An internal fan on the rotor circulates warm internal air through the internal ducts, the walls of which are cooled by the external fan blowing larger volumes of cool air through the external ducts.—Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

Solenoid Refrigerant Valve Has Removable Seat



No. 1237—New Model 73-RB solenoid valve is designed especially for liquid and suction line control of "Freon," methyl chloride, and sulphur dioxide in refrigeration and air conditioning applications.

One of its features is the removable seat assembly. By removing the bottom plug, the seat can be unscrewed, and both seat and needle assembly fall out for inspection and cleaning. It is therefore never necessary to remove the valve from the line.

The valve has ⅜ in. I.P.S. female inlet and outlet connections. The coil wire opening is tapped to receive ½ in. standard conduit or BX fittings.

Other features are brass forged body, non-magnetic special alloy seat and stainless steel valve stem, completely sealed coil, impact type plunger, sturdy cover and 38 in. lead-in wires.—Automatic Products Co., 2450 N. 32nd St., Milwaukee, Wis.

Washer Punch Has Many Uses

No. 1238—New washer punch for general use will soon pay for itself in time and labor saved, says the maker, by providing a means to produce quickly any washers that may be needed for a particular job. The 18 hardened dies ranging from ⅛ to 1⅜ in. make it possible to obtain over 150 different washer sizes.

The dies are mounted in an eccentric turret plate making it convenient to center and withdraw the washers from the machine. This punch is intended for use with fiber, steam packing, straw board, asbestos, leather, felt, cork, rubber, bakelite, mica, and other similar materials up to 1/16 in. thickness.—Ideal Commutator Dresser Co., 1537 Park Ave., Sycamore, Ill.

FOR INSULATING DUCTS

Efficiently—

Quickly . . . and Economically

SPECIFY

CELOTEX

Reg. U. S. Pat. Off.

FOR solving air conditioning duct insulating problems—quickly and effectively—engineers everywhere are specifying Celotex.

Celotex is the big, lightweight, rigid, cane fibre insulating board that handles easily, goes up fast—stays put. And it permits the maintenance of desired temperatures economically, because it *fits tight* and has a thermal conductivity of only 0.33 Btu. per inch thickness.

For insulating cold air ducts, be sure to specify Celotex VLTI—the vaporproofed low temperature insulating blocks that are *completely sealed* against vapor penetration at our factory by a special wrapping and coating. They *keep out* moisture—*keep down* operating costs. The conductivity of Celotex VLTI is 0.30 Btu. per inch thickness.

Both Celotex Insulating Cane Fibre Board and Celotex VLTI are protected against termites and dry rot by the *exclusive* Ferox Process (patented). For all the facts about these widely used insulations—write us today.

THE CELOTEX CORPORATION • 919 N. MICHIGAN AVE., CHICAGO, ILL.

CELOTEX

BRAND—INSULATING CANE BOARD

Reg. U. S. Pat. Off.

World's Largest Manufacturer of Structural Insulation

Now an...

AUDITORIUM LICENSEE

Westinghouse—one of the world's greatest engineering organizations—joins with the other notable concerns listed below, as Licensees under Auditorium Patents. Westinghouse Electric & Manufacturing Company is authorized to install, or grant permission to others to install, Central Station Air Conditioning Systems embodying Auditorium inventions. It also has the right to manufacture and sell Unit Conditioners utilizing these same patents.

AMERICAN BLOWER CORPORATION—Detroit, Mich.
BUFFALO FORGE COMPANY—Buffalo, New York
CARRIER CORPORATION—Newark, New Jersey
FRICK COMPANY—Waynesboro, Pa.
GENERAL ELECTRIC COMPANY—Schenectady, N. Y.
J. O. ROSS ENGINEERING CORP.—New York, N. Y.
THE COOLING & AIR CONDITIONING CORP.
(Division of B. F. Sturtevant Co.)
Hyde Park, Boston, Mass.
WESTINGHOUSE ELECTRIC & MANUFACTURING COMPANY—East Pittsburgh, Pa.
YORK ICE MACHINERY CORPORATION—York, Pa.

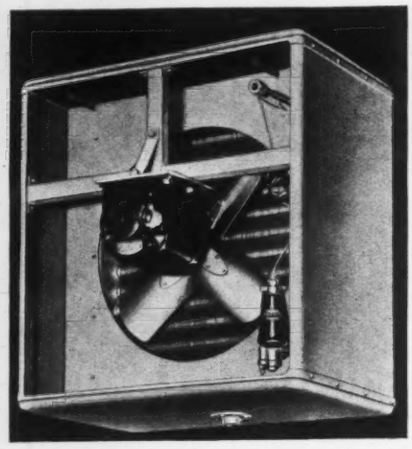
Any of the Licensees above, their Branch Offices, Distributors, or Dealers can furnish information and diagrams describing Auditorium Systems. For details of the Auditorium Plan you may write direct to

AUDITORIUM CONDITIONING CORPORATION

New York Office — 17 East 42nd St., New York

Unit Coolers for Comfort Cooling, Commercial Refrigeration

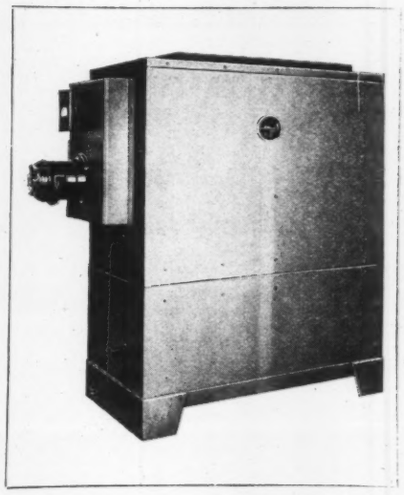
No. 1239—New "Series 73" unit coolers have been designed for comfort cooling and commercial refrigeration work. They are complete with thermostatic expansion valve, motor, and fan and are ready to connect to a remote condensing unit. Models are available for capacities ranging from 4365 to 9400 Btu per hr for refrigeration work and as comfort cooling conditioners, capacities in Btu per hr range from 17,180 to 33,300.



The cooling element is made up of 5/8 in O. D. seamless copper tubing with flat copper fins arranged vertically to allow quick drainage of condensate. Inlet and outlet of all banks are manifolded in parallel by means of headers brazed to the tubing. The cabinet is insulated at the bottom to prevent accumulation of condensate on the outside. The units are built for use with all standard refrigerants except ammonia.—Fedders Mfg. Co, Inc., 57 Tonawanda St., Buffalo, N. Y.

Announces Conditioning Units

No. 1240—New units designed to provide winter and summer air conditioning are manufactured in two types, one for commercial and large residential application and the other for domestic use. Both can be used with either steam or hot water.



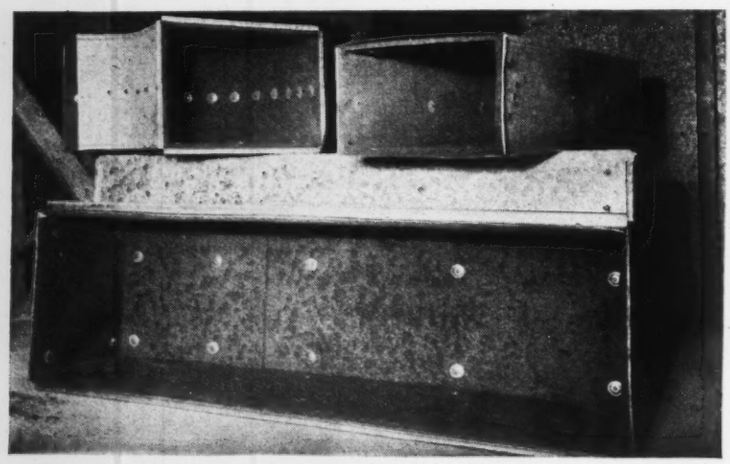
The standard commercial type model is designed for floor mounting, but can be had for ceiling suspension. It is made in three sizes to provide heating capacities ranging from 139,000 to 346,000 Btu per hr, cooling capacities from 4 to 15 tons and an air circulation of 2,000 to 5,500 cfm. Both floor and ceiling models are made up of two sections, one consisting of a dehumidifier and drain pan and the other the heating coil and fan. Humidification is spray type, provided by "Kant Klog" spray heads in dehumidifier base. Units are finished in corrosion resistant gray baked enamel.—Herman Nelson Corp., 1824 Third Ave., Moline, Ill.

Dial Thermometer Adjustable to Any Angle

No. 1241—New dial thermometer is adjustable to any angle in relation to the stem and has been designed to prove an easier means of reading important temperatures. It may be substituted for an ordinary glass thermometer as its bulb will fit the socket used for the glass thermometer. Two dial sizes are available, 4½ and 6 in. The movement, capillary, bulb and union bushing are of stainless steel in the "Dura" grade and in the "Quality"

LINING DUCTS for Sound Absorption?

...Make sure you get FIRE and MOISTURE RESISTANCE, too!



REPORTS of fires taking hold on combustible duct linings and causing serious damage come as a warning from the National Board of Fire Underwriters!*

With life and property in danger, the use of a material that provides fire-safety, as well as sound absorption, becomes a necessity.

J-M Air Acoustic Sheets perform exactly that function. This sound-absorption material cannot burn or smolder . . . will never contribute to fire spread. And being moisture resistant, Air Acoustic Sheets are exceptionally free from the dangers of humidity.

Furthermore—their high rate of sound absorption (up to 80%) . . . light weight, assuring easy handling and application . . . the permanence of their mineral composition . . . their high transverse strength and low thermal conductivity . . . all make this material outstanding in its field. For complete data on J-M Air Acoustic Sheets, write Johns-Manville, 22 E. 40th Street, New York City.

*A recent bulletin from the N.B.F.U. reports several cases in which combustible duct linings caught fire and contributed to greatly increased loss.

Johns-Manville
AIR ACOUSTIC SHEETS FOR DUCT LINING

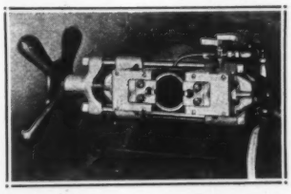
REX-WELD "Super-Service" VIBRATION ABSORBERS

for Air-Conditioning Machinery

● Here's the ideal solution to the compressor noise problem! Installed in the suction and discharge lines, these Rex-Weld Flexible Metal Tubing Units effectively dampen the vibration and put an end to the annoying hammering—surely, safely and economically . . . Produced by an exclusive patented process which forms the flexible tubing corrugations without weakening the special, non-porous, non-crystallizing bronze alloy. Supplied complete with capped ends ready for sweating onto standard water tubing. Write for Bulletin RW-4 and Data Sheet. Send us your specifications for prices.

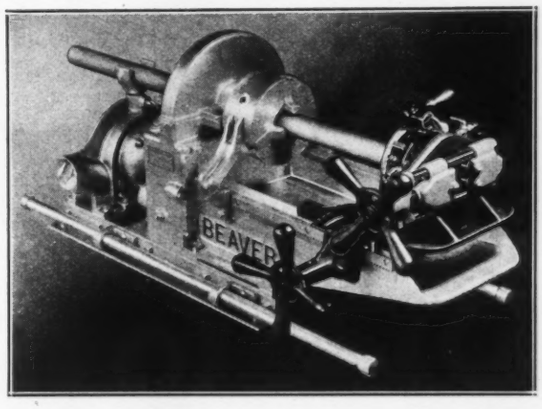
CHICAGO METAL HOSE CORPORATION
(Formerly Chicago Tubing and Braiding Company—Est. 1902)
Maywood, Illinois (Chicago Suburb)

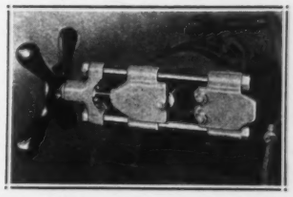
Your Choice—and at the Same Price!



Knife Cutoff

The automatic-feed knife cutoff, interchangeable with the wheel cutter by removing one set screw, is preferable for cutting soft electric conduit. It is necessary for "beveling" pipe for welding and for "grooving" pipe for Victaulic joints—the latter use requiring special knives.





Wheel Cutoff

The wheel cutter renders "double" utility—since it will cut off both pipe and solid round bars. Moreover, it is simple, fast and fool-proof. A single cutting wheel costing only a few cents will give months of service. The slight burr is instantly removed by a swinging cone reamer.

The BEAVER Model-A which led the way with such important improvements in small pipe machines as (a) rack-and-pinion feed (b) quick-opening adjustable dieheads, (c) full ⅛ to 2-inch range, (d) double-duty pipe and bolts, (e) right hand operation—now establishes another new precedent—the choice of interchangeable wheel-and-roller or automatic feed knife cutoff at the same price. Either of these cutters can be removed and used right up to the chuck for cutting short lengths—a patented Beaver advantage.

Write for New Complete Pipe Machine Catalog!

BEAVER PIPE TOOLS

637 Keeney Ave. **The Quality Line—Since 1900** **WARREN, OHIO**

MODEL W-300
Heavy duty, four cylinder, water cooled condensing unit.

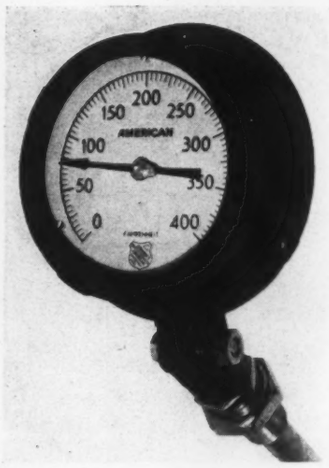
EXPECT *Dependable* STRENGTH IN THIS LINK OF THE "CHAIN"

An air conditioning system is only as dependable as its weakest link! Make sure the condensing unit YOU specify measures up in mechanical stamina with the other links... that it is geared up to economical year-in-year-out service. Brunner Condensing Units, backed by over a quarter century of sound engineering and manufacturing experience, measure up to today's exacting air conditioning requirements... A broad range of 1937 models for all installations up to 15 tons of refrigeration. Brunner Manufacturing Co., Utica, N. Y., U. S. A.

BRUNNER
Condensing Units

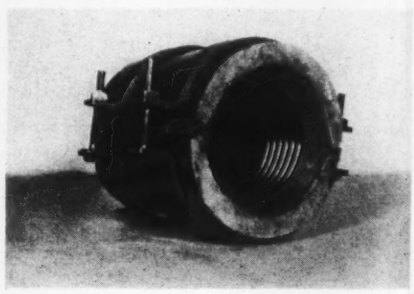
grade, the movement is of nickel plated bronze with Monel bushings and the union bushing is bronze or stainless steel. A type is available with a separable socket for use on pipes or tanks with or without lagging.

The manufacturer states that the thermometer is accurate within 1 per cent of the scale range.—American Schaeffer & Budenberg Div., Consolidated Ashcroft Hancock Co., Inc., 11 Elias St., Bridgeport, Conn.



Split Sleeves Available for Stoneware Piping

No. 1242—A full line of "Flexlock" split sleeves has been made available for use with chemical stoneware piping. They serve three main purposes: (1) they may be used to perform the same functions as unions in a screwed joint line; (2) they may be used to insert a new fitting or new length of pipe in an old pipe line which has been broken; (3) they may be used at every joint or at different intervals along the line where pipe lines must be completely or partially taken down from time to time.

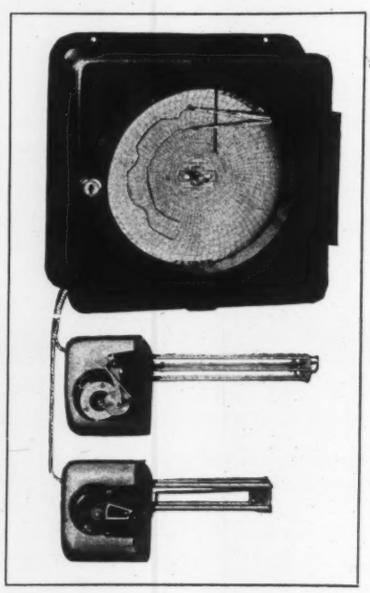


They have the advantage that twice the joint angularity attainable with standard rubber joints is permitted. Sizes from 1½ in. and larger are available.—U. S. Stoneware Co., 60 E. 42nd St., New York, N. Y.

Recording Instruments for Air Conditioning

No. 1243—New round chart recording instruments for relative humidity and temperature are available in several forms for various types of applications.

Recorders for either daily or weekly operation are available. The standard instrument has an electric clock movement, but spring wound clocks are available on special order. All recorders will handle the complete relative humidity range from 0 to 100 per cent. Various temperature ranges may also be covered; the standard is from 0 to 100 F.



One type of this new recorder transmits relative humidity direct from any remote distance, with precision and reliability. Temperature is also transmitted from remote distances. The record shows the relative humidity and temperature condition in

WANT A BARGAIN IN ERMINE?

WHAT an ermine coat is to lesser furs—Ric-wiL Certified Dry-paC Insulation is to ordinary pipe coverings. With one great exception—Dry-paC, for what it is and does, is low-priced! No pipe insulation, packed or sectional, is worth putting into underground steam line construction on price alone! Dry-paC is so far ahead of cheaper materials that most engineers see the point at once. Combined with other basic features of Ric-wiL design, Dry-paC puts Ric-wiL Conduit in a class by itself.

Prevent leakage and loss in your steam lines, keep them tight, dry and 90% efficient with a Ric-wiL Tile or Cast Iron System. Ric-wiL service is complete from design to installation and tests, and is nation-wide. On any sub-soil steam conveyance problem—under-supply, fuel waste, loss in transit—Ric-wiL can give you effective assistance. Write at once for complete catalog and test data.

The Ric-wiL Co. Union Trust Bldg. Cleveland, Ohio
New York San Francisco Chicago
Agents in Principal Cities

Dry-paC is pure, unadulterated long fibre asbestos, needing no artificial elements, no binder. Absolutely *open* to inspection during installation.

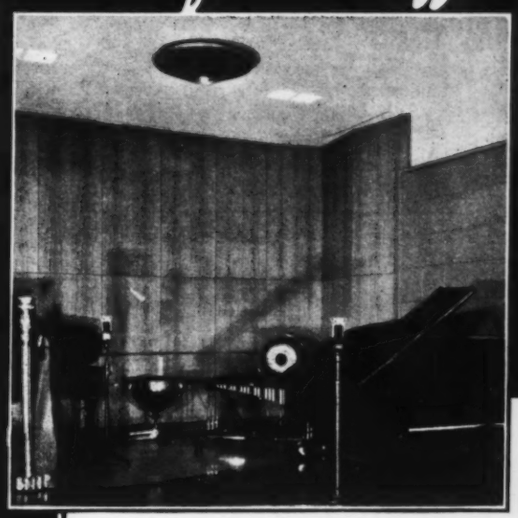
REGISTERED IN U. S. PATENT OFFICE

RIC-WIL

CONDUIT SYSTEMS FOR UNDERGROUND STEAM PIPES

ANEMOSTAT

Gives Draftless Air Diffusion



Studio of the
Columbia Broadcasting System, Inc.

The artists of the Columbia Broadcasting System perform in perfect cool comfort without experiencing the harmful and unpleasant sensation of draft.

Installation Data:

Length of Room	—38′
Width of Room	—21′ 6″
Height of Ceiling	—13′ 6″
Velocity in Duct and Anemostat Neck	—900 f. p. m.
Temperature Differential	—15°
Total Volume of Air	—2500 c. f. m.
Number of Anemostats	—2
Each Anemostat Distributes	—1250 c. f. m.

The ANEMOSTAT is an accepted scientific achievement. It has solved major problems of the Air Conditioning Industry by reducing the speed of air flow to such an extent that it is no longer perceptible as a draft and by mixing room air with conditioned air within the device. Its use, therefore, permits greater temperature differentials, smaller ducts, smaller plants, lower installation and operating costs. The ANEMOSTAT can be applied to existing systems as well as to new installations.

"No Air Conditioning System is better than its Air Distribution"

ANEMOSTAT CORP. of AMERICA

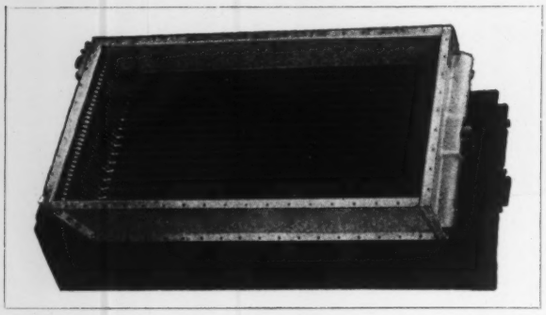
551 FIFTH AVE. NEW YORK CITY.

a space where the sensitive elements are located, while the recorder itself may be located at any suitable location, practically any distance away. The only connection required between the humidity and temperature sensitive elements is a low voltage 5 or 8-conductor wire.

The illustration shows the recording instrument itself and also the sensitive electric transmitting element which is employed in the distant transmission type.—Julien P. Friez & Sons, Inc., 4 N. Central Ave., Baltimore, Md.

New Blast Heater Developed

No. 1244—A new blast heater for heating, ventilating, air conditioning and process application incorporates new features of design which materially increase structural strength and make for more effective heat transfer, according to the manufacturer.



Among the features is elimination of expansion strain. The expansion bend allows each tube to expand and contract as its temperature requires, without affecting the tube adjacent to it. Headers and tubes are cylindrical and seamless and are brazed into a single unit without use of gaskets, bolts or screw joints.

All steam carrying passages of the blast heater condenser, including headers, tubes, and inlet and outlet bosses are of pure copper or copper alloy from the point where the steam enters to where it leaves in the form of condensate. Orificing of tubes has been eliminated by use of a special steam distributing plate which rations steam evenly to all tubes the full width of the condenser.

To closer approximate final temperature and physical size requirements, this blast heater is available in a large variety of sizes and capacities, a total of 252 different heaters.—Modine Mfg. Co., 17th St., Racine, Wis.

One Piece Welding Helmet

No. 1245—A welding helmet of entirely new design is formed of one piece black vulcanized fiber cut deeply to offer side protection to a point well back of the ears, and is said to have great ruggedness to withstand hard shop use.

The smooth, rounded interior promotes air circulation and results in a well ventilated, cool, comfortable helmet, states the maker.—Willson Products, Inc., Reading, Pa.

High Pressure Air Eliminator

No. 1246—High pressure air eliminator has been designed to automatically vent driers, steam coils, traps, etc., otherwise vented by hand and operating under varying steam pressures up to 150 lb.

The operating member is made of a non-corrosive metal sensitive to heat changes, which closes the valve at 212 F and keeps it closed until the temperature drops below that point, when it automatically opens the valve. The body is made of bronze, nickel-plated. The air outlet is 3/16 in. in diameter.—Gorton Heating Corp., Cranford, N. J.

Relay Transformer Sets for Air Conditioning

No. 1247—A new type of relay transformer set was recently announced. The illustration shows one of these new instru-

THEY produce a finer, more uniform spray and operate efficiently under low pressures. Result, more Marley patented spray nozzles are sold than any other kind.
THE MARLEY COMPANY, Kansas City, Mo.
MARLEY
PATENTED SPRAY NOZZLES

LET'S STOP THESE "Hold-ups"

Every year, corrosion — the greatest racketeer in the world — robs this country of millions of dollars. Fortunately for pipe users, a large portion of this money can be saved — by using Fretz-Moon Copper-Bearing Steel Pipe. • When you are confronted with alternate wet and dry conditions — atmospheric corrosion — and steel pipe doesn't last as long as you or the customer would like, tell him about this rust-resisting pipe made of high quality steel with which is alloyed the correct amount of copper to obtain maximum corrosion-resistance. It provides the high strength of steel *plus* the exceptional uniformity of product — size, ductility and cleanliness — made possible by the patented "continuous process" employed in manufacture. And the additional cost will be returned many times over through the longer life it affords. • Ask your jobber for Fretz-Moon Copper-Bearing Pipe, black or galvanized. If he doesn't stock it, write us.

FRETZ-MOON TUBE CO., INC.
BUTLER • PENNA.

FRETZ COPPER MOON BEARING

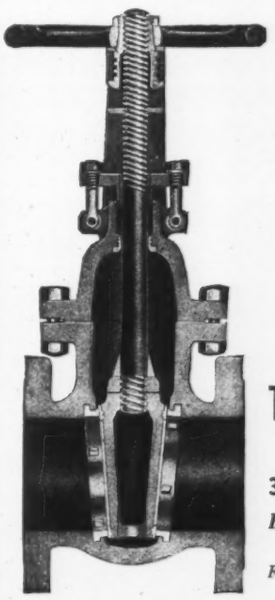
STEEL PIPE

Do You Discard Shoes When the Soles Wear?

It is even more wasteful to scrap an entire valve simply because the seat is worn. Yet that is what is usually done with ordinary valves.

When the seats of a Fairbanks Valve wear, the valve can be made as good as new in a few minutes, at small cost, by replacing the worn seat rings with new ones. It is not even necessary to remove the valve from the pipe line.

Valve users throughout the Nation have been amazed at the savings effected by the above and other Fairbanks Valve features. Try Fairbanks "Renewables" and see how much they will reduce your valve maintenance costs.



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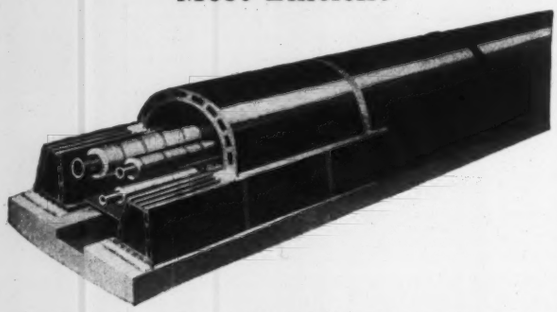
Boston, Pittsburgh — Distributors in Principal Cities
Factories: Binghamton, N. Y. Rome, Ga.

Fairbanks Renewable Valves

THERM-O-TILE

The Conduit for Underground Steam Lines

Simplest
Strongest
Most Efficient



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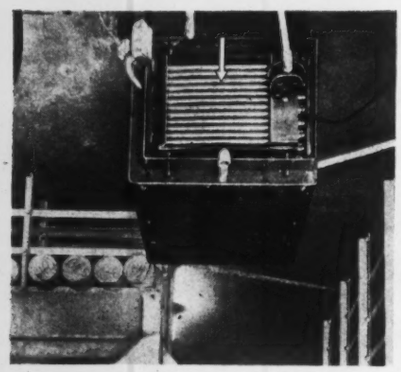
CHROMALOX
ELECTRIC HEATING UNITS



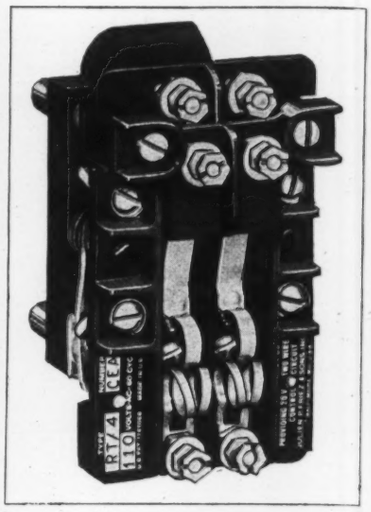
enable heating contractors to meet difficult conditions

For instance take the case illustrated above. To "condition" concrete samples a cement company desired a room in which the temperature was always uniform, and 100 percent of humidity always maintained. Water sprays, a unit type cooler and a Chromalox 5 Kw. electric air blast heater made up the equipment, all controlled automatically. Due to the high humidity, water proof design was essential. The air blast heater met this need completely, and automatic temperature control insures uniform temperature.

The Chromalox air blast heater is only one of a line of electric heating units which enable the heating contractor to broaden his scope, embracing a wide range of industrial heating. Get the facts, in the Chromalox Book of Electric Heat (on request), and bring your electric heat problems to Wiegand engineers.

EDWIN L. WIEGAND COMPANY
7610 Thomas Blvd. Pittsburgh, Pa.

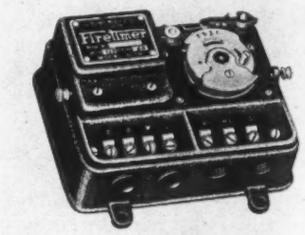
ments mounted in its housing.



These sets depend upon a recent discovery and are novel in design. Because of this feature it is possible to obtain relays for handling appreciable electrical loads (up to ¾ hp or 1000 watts), and also at the same time a transformer at a cost no higher than for a simple relay alone, according to the maker. In this way, the electrical load to be handled remains on line voltage (which may be either 110 or 220 volts) while the control instrument (thermostat, humidistat, or effective temperature control) is at the same time placed on low voltage. The same coils which operate the relay itself also produce the low voltage supply (20 volts).—Julien P. Friez & Sons, Inc., 4 N. Central Ave., Baltimore, Md

New Stoker Features Announced

No. 1248—The "Fire-Timer," improved control for automatic coal burner operation illustrated, and a shear pin alarm are new stoker features announced last month. The timer prevents losing fire during mild spring or fall weather, by automatically operating the stoker for brief periods at predetermined intervals. Although independent of the thermostatic control, the timer is co-ordinated with it to avoid excessive burning.

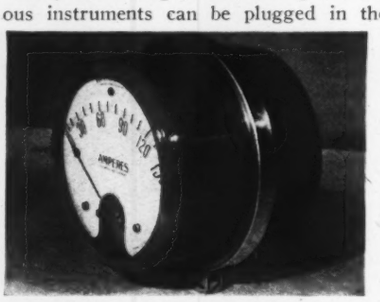


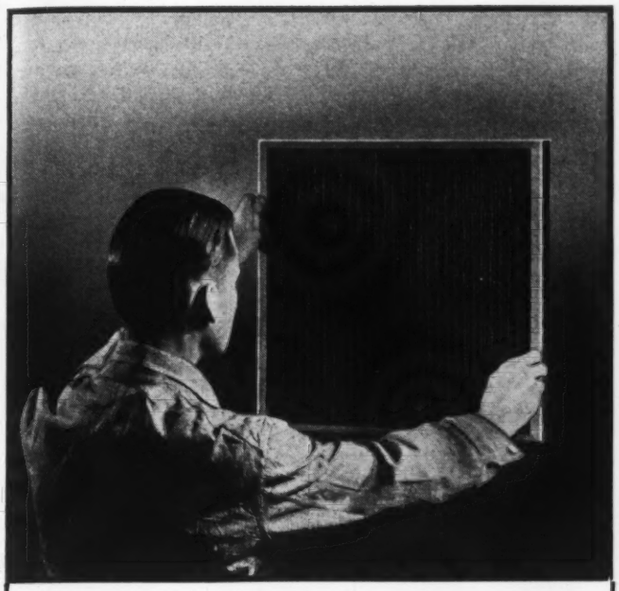
The shear pin alarm gives suitable warning when some foreign object has been carried in with the coal and has caused the safety pin to shear before any damage has been done. The alarm is electrically operated and the warning may be in the form of a flashing light, a buzzer or bell, etc.—Fairbanks, Morse & Co., 900 S. Wabash Ave., Chicago, Ill.

Plug-In Instruments Need No Panel

No. 1249—New low cost detachable instruments for general industrial use whose sockets may be cut into the conduit run feeding a motor or grouped in standard metal boxes to constitute a panel assembly were announced last month.

Sockets can be installed in circuits feeding electrical loads in industrial plants and processes where it would be helpful to know consumption and performance characteristics, even if only occasionally. Advantages are: (1) socket provides its own switchboard, eliminating panels, wiring and mounting details; (2) various instruments can be plugged in the same socket to obtain volts, amperes, watts, power factor, kilowatt-hours, etc.; (3) sockets can be installed and sealed off providing convenient outlets for future installation of instruments or making connections for portable analyzers for periodic testing.





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Exacting buyers demand the exclusive advantages offered only by Air-Maze. Write for Bulletin BPAN-86.

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- 99.3%-99.8% Efficient
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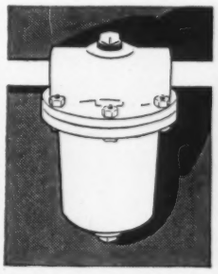
AIR-MAZE
CLEANABLE AIR FILTER PANELS

THESE FITTINGS LEFT OVER IN REPLACING OLD TRAPS WITH ANDERSON SUPER-SILVERTOPS

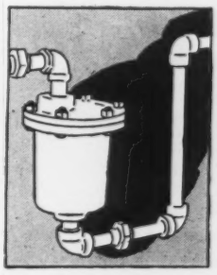
RECENTLY when Anderson Super-Silvertop traps replaced all the old traps in a process plant, this pile of fittings was left over. No new fittings were needed but 988 miscellaneous fittings were of no further use. This is just one instance that shows the saving in fittings when Anderson Super-Silvertop simplified piping is specified.

Other advantages realized by using Anderson Super-Silvertops are; Trap is inspected without disturbing pipe connections. The extra capacity of Super-Silvertop permits the use of smaller, less expensive traps. Anderson Super-Silvertop traps make a more attractive appearance —the piping is simplified and clean cut. Find out more about Anderson Super-Silvertop, the only complete steam trap. Send for full information and prices.



At Left: Super-Silvertop trap installed straight-in-line. Can also be installed as elbow.

At Right: How some of the old traps were installed.



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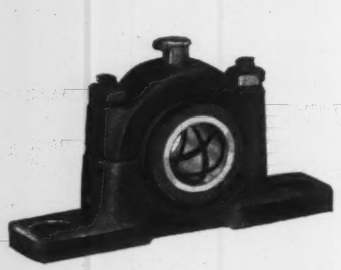

Randall Standard Pillow Block assembled. (Patented)

Cut-away section of ball of Randall Standard Pillow Block

Randall Pillow Blocks will last practically a lifetime with a minimum of attention. They are self-aligning, self-lubricating and are clean and quiet in operation.

On more than 75% of all air conditioning units produced, Randall Pillow Blocks are standard equipment.

Write for new catalog illustrating and describing our full line of pillow blocks.

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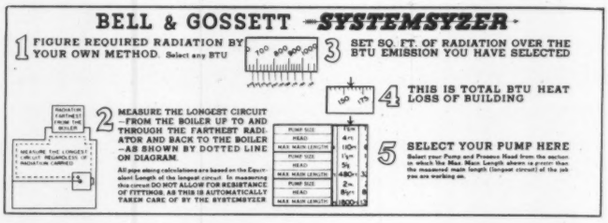
New descriptive bulletins just off the press on McQuay Blast Coils, Unit Heater, Unit Coolers, Cabinet Radiation, Cabinet Coils, Concealed Radiation, etc.

McQUAY, Inc.
Minneapolis, Minn.

A detachable instrument consists of two main units, first a socket or mounting device which is provided with electrical connection jaws, and second, the instrument mechanism mounted on a plate having electrical connection blades on the back and enclosed in a weatherproof housing of glass.—Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

New Slide Rule Simplifies Forced Hot Water Heat Design

No. 1250—A new slide rule that simplifies designing and pipe-sizing a forced hot water heating system is called the "B & G Systemsyzer." With it, the heating engineer or contractor need only figure, by the usual method, the amount of radiation required and measure the length of the longest main circuit, as laid out on the plan. With these figures known, the rule takes up the job and completes the necessary calculations.


BELL & GOSSETT SYSTEMSYZER
1 FIGURE REQUIRED RADIATION BY YOUR OWN METHOD. Select any BTU
2 MEASURE THE LONGEST CIRCUIT —FROM THE BOILER UP TO AND THROUGH THE FARTHEST RADIATOR AND BACK TO THE BOILER —AS SHOWN BY DOTTED LINE ON DIAGRAM.
3 SET SQ. FT. OF RADIATION OVER THE BTU EMISSION YOU HAVE SELECTED
4 THIS IS TOTAL BTU HEAT LOSS OF BUILDING
5 SELECT YOUR PUMP HERE

For example, the first calculation in forced hot water designing is to determine the pump size and the proper pressure head. Using the device, the designer is given both figures by making one move of the slide. Sizing the pipe, for both one and two pipe systems, is an equally simple operation.—Bell & Gossett Co., 3000 Wallace St., Chicago, Ill.

Start-Stop Control and Unloader for Compressors

No. 1251—Automatic start-and-stop control, combined with automatic loading and unloading, can now be obtained without additional equipment on motor driven, single horizontal air compressors, by using a new magnetic unloader. No extra piping, valves or other devices are required.

This device operates on standard electric equipment and can be installed wherever a conventional automatic starter is used for controlling the compressor motor. The only other provision necessary is a pressure switch to operate a pilot circuit. The pressure switch control is transferred electrically from the motor starter to the magnetic unloader, and either position may be selected at will. If connected to the motor starter, the compressor starts and stops on the demand for air. When connected to the magnetic unloader, the compressor runs continuously and simply loads and unloads depending upon whether or not air is needed. If desired, the cooling water supply can also be automatically controlled.—Worthington Pump and Machinery Corp., Harrison, N. J.

Becomes Auditorium Licensee

It was announced last month that Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa., has become a general licensee under Auditorium patents and is authorized to install, or grant permission to others to install, central station air conditioning systems embodying Auditorium inventions. It also has the right to manufacture and sell unit conditioners utilizing these same patents.

Other general licenseees include: American Blower Corp., Buffalo Forge Co., Carrier Corp., Frick Co., General Electric Co., J. O. Ross Engineering Corp. The Cooling & Air Conditioning Corp. (Div. of B. F. Sturtevant Co.) and York Ice Machinery Corp.

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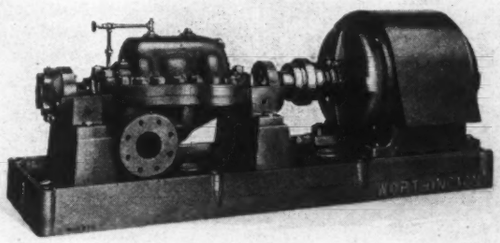
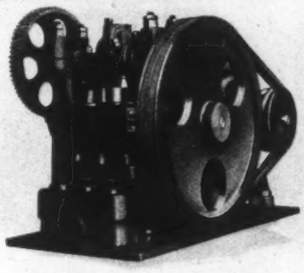
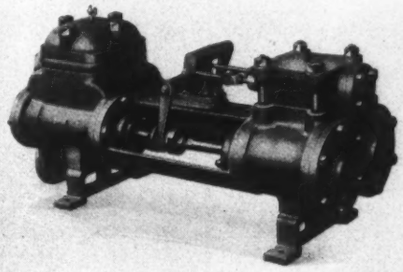


Horizontal Split-case Multi-stage Centrifugal Pump
Capacities to 2000 g. p. m. . . . Pressures to 1200 lb. per sq. in.



Vertical Triplex Power Pump
Capacities to 500 g. p. m. . . . Pressures to 2000 lb. per sq. in.



Horizontal Duplex Steam Pump
Capacities to 225 g. p. m. . . . Pressures to 300 lb. per sq. in.

THE product of unsurpassed engineering and rigid quality standards . . . they are representative of a comprehensive line, providing an exact selection for any individual condition without compromise.

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AEROFIN CLEANABLE TUBE UNITS: For cooling only. Headers removable to permit cleaning of tubes. Recommended where sediment or scale-forming chemicals are present in cooling water.

AEROFIN DIRECT EXPANSION UNITS: Centrifugal header one valve control type. Recommended where control of rows in direction of air flow is not required.

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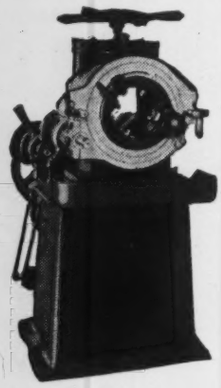

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267 5th Ave., New York City, N. Y. 421 Dwight Building, Kansas City, Mo.

Book Reviews

Mechanics of Turbulent Flow

"The Mechanics of Turbulent Flow," by Boris A. Bakhmeteff, Professor, Columbia University. Lectures delivered under the William Pierson Field Foundation at Princeton University, February, 1935. 101 pp., 6x9 in., clothbound. Published by the Princeton University Press, Princeton, N. J. Price, $2.50.

Engineers concerned with problems of fluid flow in pipe lines will welcome this series of lectures by Professor Bakhmeteff in book form. In them, he has presented recent advances in this field in a clear and interesting manner which should prove valuable in stimulating future progress.

The first chapter is concerned with fundamental notions, including energy loss and shearing stress, laminar and turbulent motion, the Reynolds number and dynamic similarity, friction in laminar motion, and *smooth* and *rough* surfaces. The second chapter deals with mechanics of turbulent flow, the discussion being divided into the following topics—momentum exchange, the mixing length, turbulent and viscous agents inside a fluid in motion, the laminar film and Prandtl's "boundary layer," distribution of mixing length, and velocity patterns.

Chapter three is devoted to Kármán's theory of the turbulent flow pattern—the dimensionless velocity deficiency law, logarithmic velocity curve law, relation between maximum and average velocities, concluding with remarks and interpretations. The fourth lecture deals with final formulas for flow in smooth and rough pipes, and the book concludes with the author's remarks on the outlook for future research.

Recent Trade Literature

For your convenience in obtaining copies of bulletins, see coupon on page 144.

No. 2382. AIR CONDITIONING. Airtemp, Inc., Leo St., Dayton, Ohio. 16 p. booklet on cooling for profit describing new summer air conditioner in detail and showing how it may be adapted to various decorative schemes in stores, shops, offices, hotels, hospitals, etc.

No. 2383. AIR CONDITIONING: Carrier Corp., 850 Frelinghuysen Ave., Newark, N. J. 16 p. booklet presenting an easy way to find out how quickly air conditioning pays for itself in theaters, and giving facts and figures based on experience in conditioning all types and sizes of theaters ranging in capacity from 15 to 6200 seats.

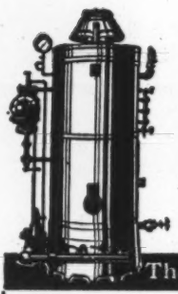
No. 2384. AIR CONDITIONING: Westinghouse Electric & Mfg. Co., Mansfield, Ohio. 12 p. booklet on how to buy air conditioning, discussing the factors involved in purchasing the type of system best suited to the installation, methods of control, and cost of operation.

No. 2385. AIR CONDITIONING AND REFRIGERATION PARTS: Airo Supply Co., 2726 N. Ashland Ave., Chicago, Ill. 86 p. wholesale catalog of refrigeration and air conditioning parts, tools, and supplies.

No. 2386. AIR COOLERS: General Electric Co., 1 River Rd., Schenectady, N. Y. 8 p. bulletin on surface air coolers for turbine generators.

No. 2387. AIR FILTERS: Air-Maze Corp., 812 Huron Rd., Cleveland, Ohio. 8 p. bulletin on industrial air filters in oil bath, multimaze, and panel types for air conditioning and blower systems, air compressors, pumping units, gas, gasoline and diesel engines, with formulas for determining sizes, and other information; also, 12 p. booklet on automotive air filters.

No. 2388. AIR HEATERS: James Campbell Smith, Inc., 16374 Euclid Ave., Cleveland, Ohio. 4 p. bulletin describing



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The KANE Automatic Gas Fired Boiler

THIS boiler won't steal valuable time from your working day!
Just start it up in the morning—adjust it to the pressure required—and forget it.
The KANE operates automatically . . . will maintain the desired pressure, unvarying, with no more than a very few minutes manual attention each day. And it will supply the absolutely *clean* steam so important to the supply for presses.
No other boiler has all the advantages of the KANE . . . and none is so widely used!
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Literature upon request.

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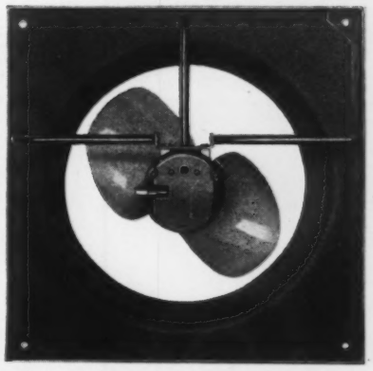

Their performance will justify all the good words you can say about them. Like the entire National Line of ventilating and air conditioning equipment, they are expertly engineered and built to deliver an extra margin of service. It will pay you to learn all about the National Line. Send for catalog.

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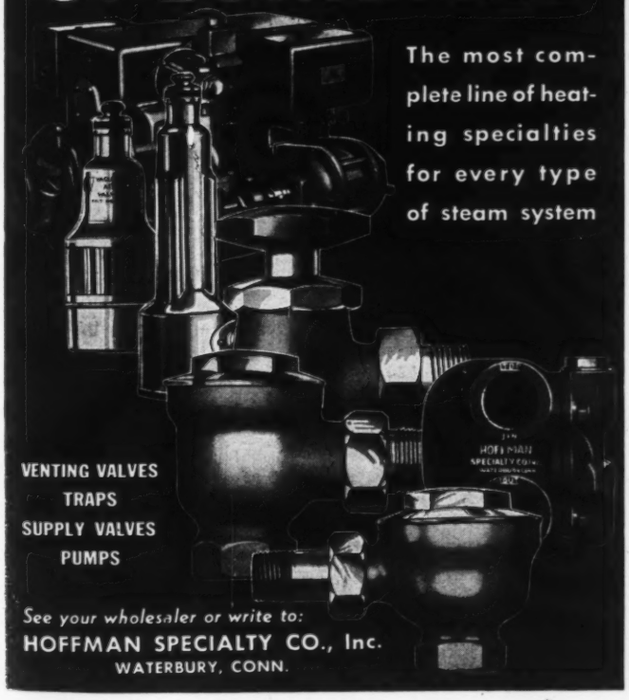

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FOR AIR WASHING

For industrial air washers we recommend either ¼" Fig. 631 (female) or ⅜" Fig. 629 (male pipe-illustrated) Brass nozzles. Size delivering 68 G.P.H. at 20 lbs. is recommended.

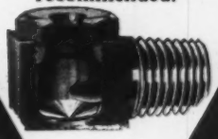

Fig. 629

For liquids with corrosive tendencies Fig. F-27 can be supplied in steel; for erosive liquids in Nitralloy.

For air conditioning work. Supplied in capacities .57 G.P.H. up @ 40 lbs. All brass nozzle.

Fig. F-27

Write for Catalog 6-A

¼" Fig. H-261

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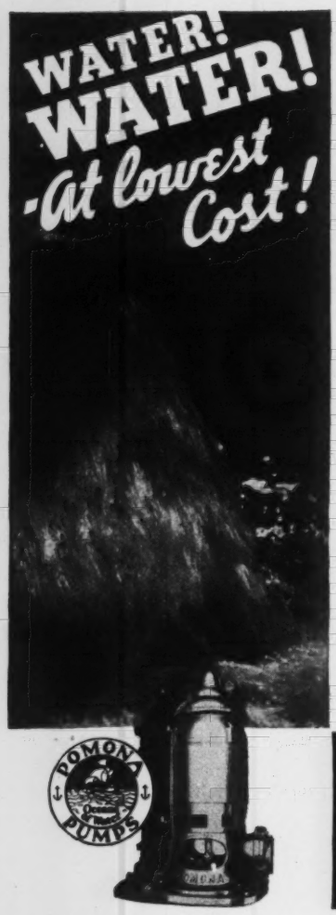

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A Pomona guarantee of efficiency is always discounted—BY US. We know that when a Pomona Pump goes to work it will perform at face value—or better. The consistent performance of Pomona Pumps up to or beyond the letter of our guarantee is a solid fact on which the preference of Pomona Pumps is based. All over the world—in all classes of service—it is the recorded performance of Pomona Pumps that sends a steadily increasing stream of new and repeat orders to our factories. Make a comparison of pump records on the basis of per gallon per year costs of water delivery. You will find that Pomona leads for economy and dependability.

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Same design features but forged from Chrome-Alloy Steel — chrome-plated. LIGHT . . THIN . . STRONG

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SUPERIOR DROP-FORGED TOOLS
ADJUSTABLE WRENCHES

NOZZLES for Air Washing

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YARNALL-WARING CO., PHILA.
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YARWAY

automatic coal fired air heater for plants, giving test data on fuel saving and details of construction and operation.

No. 2389. ALLOYS: Climax Molybdenum Co., 500 Fifth Ave., New York, N. Y. 4 p. pamphlet on how molybdenum iron helps preserve accuracy of liquid meters.

No. 2390. ALLOYS: International Nickel Co., Inc., 67 Wall St., New York, N. Y. 16 p. technical bulletin on machining Monel, nickel, and Inconel.

No. 2391. BOILERS: American Radiator Co., 40 W. 40th St., New York, N. Y. 6 p. bulletin giving ratings and data for "Ideal" oil burning boilers, for handy reference and quick selection.

No. 2392. BOILERS: Hart & Crouse Co., Inc., 301 Turner St., Utica, N. Y. 6 p. bulletin on "Royal" push nipple heating boilers with description of features and ratings and dimension data.

No. 2393. CO_2 METERS: Brown Instrument Co., Div. of Minneapolis-Honeywell Regulator Co., 4534 Wayne Ave., Philadelphia, Pa. 24 p. catalog on electric CO_2 meters for indicating and recording the percentage of CO_2, discussing how waste occurs in combustion and the use of such meters.

No. 2394. COMPRESSORS: Frick Co., Waynesboro, Pa. 12 p. bulletin on ammonia booster compressors for low temperature refrigeration, giving dimensions and other data on various sizes and discussing applications.

No. 2395. COMPRESSORS: Worthington Pump & Machinery Corp., Harrison, N. J. 8 p. condensed catalog of vertical air compressors, single stage, single cylinder, air cooled and water cooled types.

No. 2396. CONVECTORS: Trane Co., La Crosse, Wis. 8 p. issue of "Weather Magic," Vol. 1, No. 1, featuring information on convectors, evaporative condensers, unit heaters, and other equipment.

No. 2397. DRIVES: Congress Tool & Die Co., 9030 Lumpkin Ave., Detroit, Mich. 4 p. bulletin on standardized drives including grooved pulleys, variable speed pulleys, step-cone pulleys, and flexible couplings.

No. 2398. DUCT INSULATION: Grant Wilson, Inc., Taylor & Karlov Sts., Chicago, Ill. Folder describing "Dux-sulation" with a sample of the insulation material attached.

No. 2399. DUST CONTROL: W. W. Sly Mfg. Co., 4700 Train Ave., Cleveland, Ohio. 16 p. booklet on dust control discussing the requirements of an efficient dust filter, and this company's equipment and its application.

No. 2400. EXPANSION JOINTS: Foster Wheeler Corp., 165 Broadway, New York, N. Y. Folder describing stainless steel bellows expansion joints of various types for pressures up to and including 300 lb and temperatures up to 750 F.

No. 2401. FLEXIBLE METAL HOSE: Chicago Metal Hose Corp., Maywood, Ill. 26 p. manual on the applications of "Rex-weld" and "Rex-tube" flexible metal hose giving technical and application data and illustrating the design and construction of this company's product.

No. 2402. FLEXIBLE METAL TUBING: American Metal Hose Branch, American Brass Co., 414 Meadow St., Waterbury, Conn. 20 p. booklet on "American" seamless flexible metal tubing with data on construction, advantages, and application, discussing a typical installation, and illustrating and describing various types of couplings and fittings.

No. 2403. FLOW METERS: Bristol Co., Platts Bridge, Waterbury, Conn. 40 p. catalog of electrical and mechanical flow meters for recording, integrating, controlling, and indicating the flow of steam, liquids or gases.

No. 2404. FLOW METERS: Leeds & Northrup Co., 4952 Stenton Ave., Philadelphia, Pa. 16 p. catalog discussing the principle incorporated in the "Centrimax" flow meter for greater accuracy in flow integration. Test data and descriptions of equipment are given.

No. 2405. GASKETS: International Nickel Co., Inc., 67 Wall St., New York, N. Y. 4 p. folder on Monel gaskets for temperatures from 400 to 450 F and pressures from 250 to 1375

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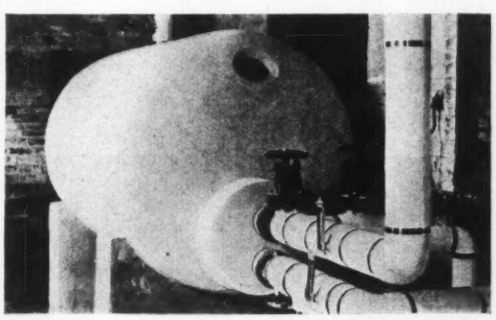

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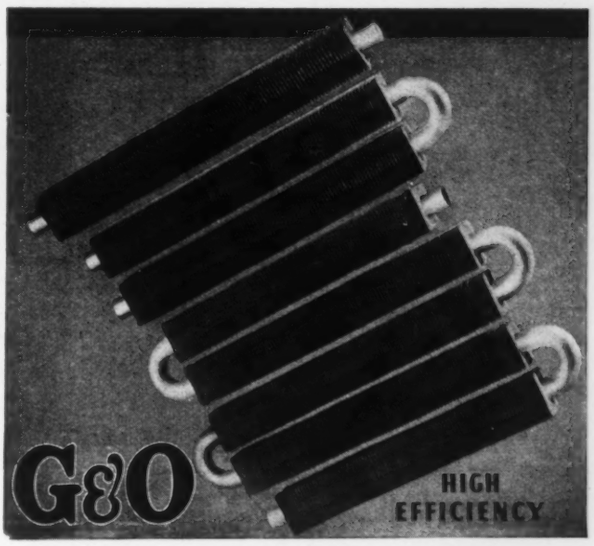

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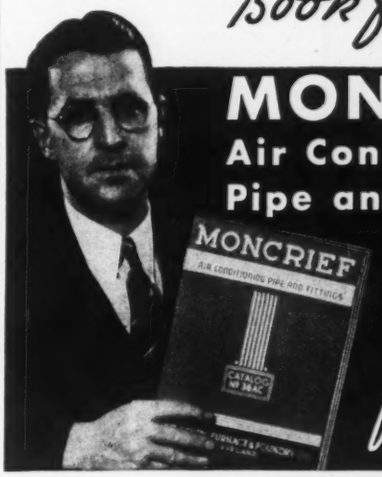

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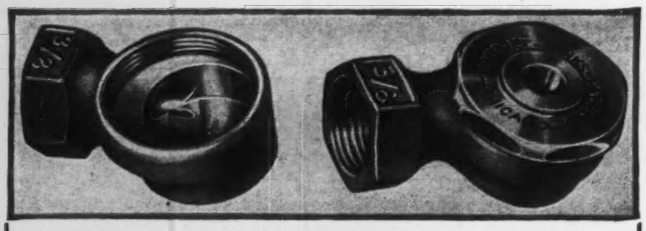

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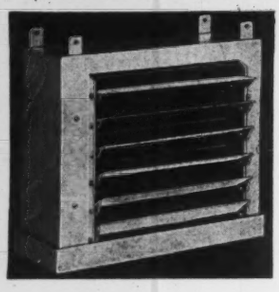

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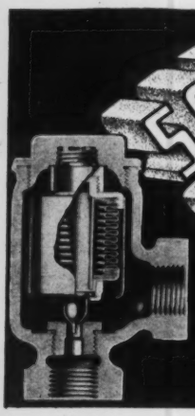

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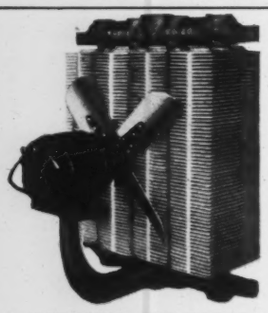

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lb per sq in. Various types are illustrated and a list of manufacturers is given.

No. 2406. HOT WATER HEATING: H. A. Thrush & Co., Peru, Ind. 12 p. booklet on the advantages of hot water heating, illustrating and describing flow control valve, water circulator, etc.

No. 2407. IGNITION TRANSFORMERS: General Electric Co., 1 River Rd., Schenectady, N. Y. 4 p. bulletin on small size ignition transformers for oil burners.

No. 2408. INSTRUMENTS: Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. Slide rule indicator for use in selecting electrical instruments for industrial plants, central stations, and the transportation industry. Shows what instruments to use, approximate price, size, and available descriptive literature.

No. 2409. MOTORS: Bodine Electric Co., 2254 W. Ohio St., Chicago, Ill. 4 p. issue of the "Motorgram" (Vol. 17, No. 2) including information on the characteristics and applications of small synchronous motors.

No. 2410. MOTORS: General Electric Co., 1 River Rd., Schenectady, N. Y. 4 p. data sheet on low speed large size synchronous motors, describing features in detail.

No. 2411. MOTORS: Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. 4 p. bulletin on dual ventilated fan cooled squirrel cage motors from 1 to 200 hp employing a heat exchanger principle of cooling.

No. 2412. PACKINGS: Metalastic Mfg. Corp., Hackensack, N. J. 16 p. catalog on this company's packings describing the various types and their applications and giving useful information.

No. 2413. PRESSURE AND VACUUM SWITCHES: General Electric Co., 1 River Rd., Schenectady, N. Y. 4 p. data sheet on pressure and vacuum switches of diaphragm type, a-c or d-c; maximum voltage 550; maximum pressures 13, 40, 80, 160, 300, and 5000 lb; maximum vacuum 28 in.

No. 2414. PUMPS: Fairbanks, Morse & Co., 900 S. Wabash Ave., Chicago, Ill. 6 p. bulletin on centrifugal fire pumps approved by the underwriters, describing features and giving principal dimensions.

No. 2415. PUMPS: Fairbanks, Morse & Co., 900 S. Wabash Ave., Chicago, Ill. 6 p. folder on simplex self oiling power pumps, describing characteristics and features and giving principal dimensions, sizes, and ratings.

No. 2416. PUMPS: Fairbanks, Morse & Co., 900 S. Wabash Ave., Chicago, Ill. 4 p. bulletin on ball bearing centrifugal pumps for high speed operation against heads up to 245 ft with capacities of from 90 to 225 gpm, describing features and giving dimensions and capacity data.

No. 2417. PUMPS: Goulds Pumps, Inc., Fall St., Seneca Falls, N. Y. 12 p. bulletin on close coupled pumps, illustrating applications, describing the various types and sizes, discussing features and advantages, and giving performance charts and dimension data.

No. 2418. PUMPS: Goulds Pumps, Inc., Fall St., Seneca Falls, N. Y. 8 p. bulletin on high pressure centrifugal pumps for boiler feeding, gasoline, and pipe line service, water works, and other high pressure service, describing features, giving specifications, and presenting size, capacity, and dimension data.

No. 2419. PUMPS: Ingersoll-Rand Co., 11 Broadway, New York, N. Y. Catalog describing "Cameron Motorpumps" for capacities from 5 to 1000 gpm and heads up to 500 ft.

No. 2420. PUMPS: International Nickel Co., Inc., 67 Wall St., New York, N. Y. 6 p. bulletin entitled "Making Old Pumps Good as New with Monel Parts" describing various examples and giving facts and data on the properties of Monel.

No. 2421. REFRIGERATION: Frick Co., Waynesboro, Pa. 8 p. bulletin on refrigeration for the oil industry including information on compressors, condensers, brine and water coolers, valves and fittings.

No. 2422. REGISTERS: Auer Register Co., 3608 Payne Ave., Cleveland, Ohio. 48 p. book on registers, comprising a

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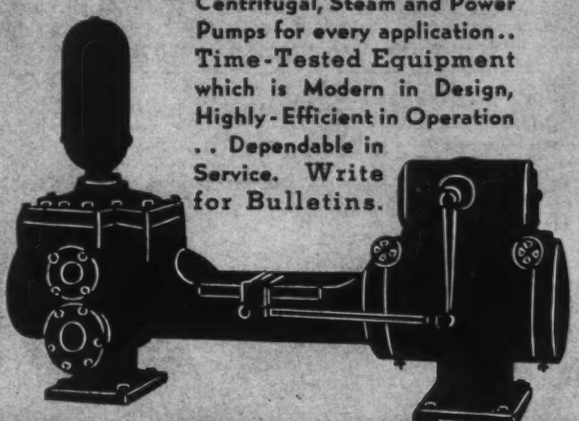

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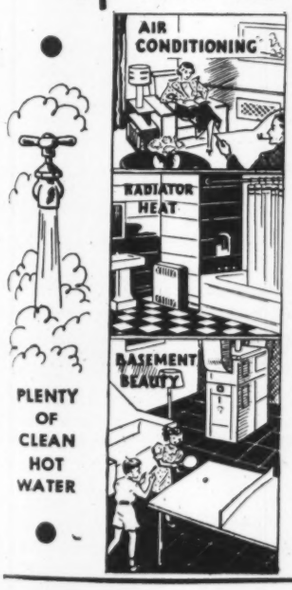

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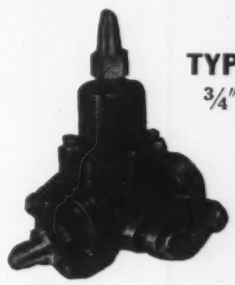

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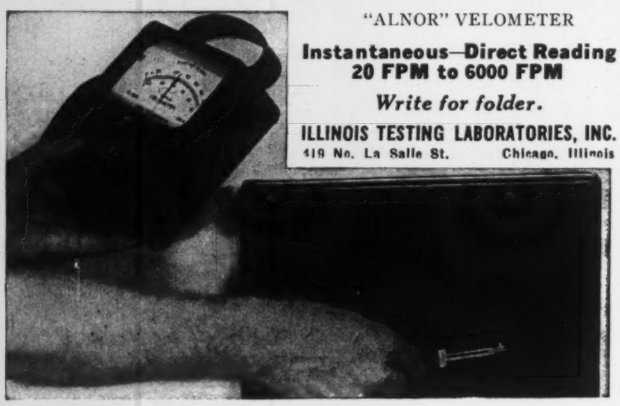

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No. 2424. SPRAY NOZZLES: Yarnall-Waring Co., 102 E. Mermaid Lane, Chestnut Hill, Philadelphia, Pa. 4 p. bulletin on "Yarway" spray nozzles for air conditioning and general industrial uses, illustrating various types and giving application data.

No. 2425. TIMERS: Automatic Temperature Control Co., 34 E. Logan St., Philadelphia, Pa. 4 p. bulletin on unusual time controls for various applications.

No. 2426. UNIT COOLERS: Perfex Corp., 415 W. Oklahoma Pl., Milwaukee, Wis. 8 p. bulletin on unit coolers describing features of design and giving capacity tables. Information on using this company's unit heaters as unit coolers, with tables of cooling capacities, is presented.

No. 2427. UNIT COOLERS: Rempe Co., 340 N. Sacramento Blvd., Chicago, Ill. 8 p. bulletin and price list of unit coolers giving dimensions, ratings, and other data. Unit coolers for comfort cooling are included.

No. 2428. VALVES: Homestead Valve Mfg. Co., P. O. Box H, Coraopolis, Pa. 24 p. pocket-size booklet on quarter turn valves discussing features and giving dimensions, price list, tables, and information on uses.

No. 2429. VALVES: International Nickel Co., Inc., 67 Wall St., New York, N. Y. 4 p. bulletin on the advantages of Monel metal trim for valves.

No. 2430. WELDERS: Harnischfeger Corp., 440 W. National Ave., Milwaukee, Wis. Booklet entitled "Arc Welding of Tomorrow" describing "Smootharc" welders of various types.

No. 2431. WELDING CABLE CONNECTIONS: Tweco Products Co., 829 S. Madison, Wichita, Kans. 4 p. bulletin on cable lugs, ground clamps, electrode holders, and cable connectors.

No. 2432. WELDING STAINLESS STEEL: Republic Steel Corp., Republic Bldg., Cleveland, Ohio. 20 p. booklet on welding of "Enduro" stainless steels, describing the proper methods for welding by electric arc, gas, seam, spot, projection, and atomic hydrogen methods. Information on brazing and silver soldering, and a table showing in condensed form the properties of 13 stainless steels is presented.